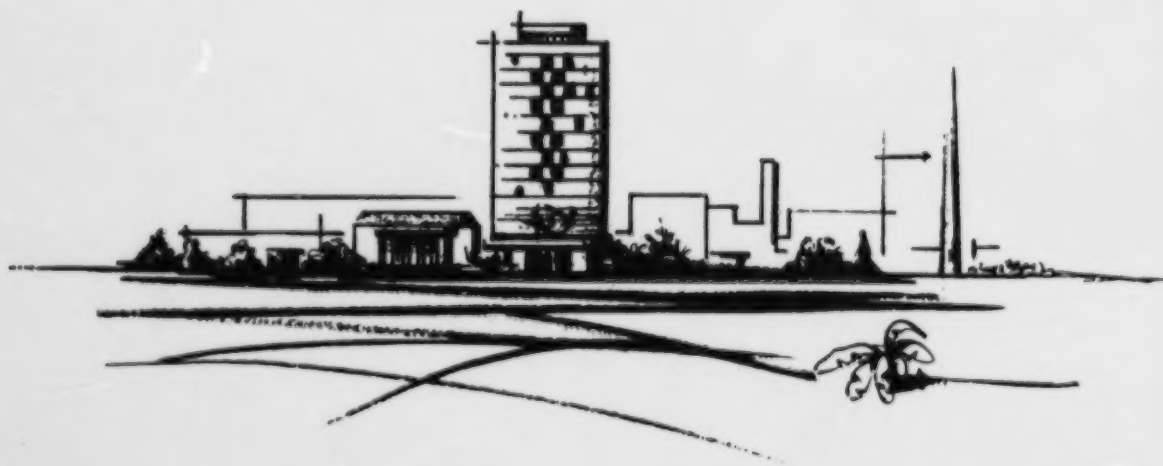


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RESEARCH REPORT



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PHASE REPORT

on

**A STUDY OF THE INFLUENCE OF FUEL ATOMIZATION,
VAPORIZATION, AND MIXING PROCESSES ON POLLUTANT
EMISSIONS FROM MOTOR-VEHICLE POWERPLANTS**

to

**Division of Motor Vehicle Research and Development
NATIONAL AIR POLLUTION CONTROL ADMINISTRATION**

Contract No. CPA 22-69-9

April 30, 1969

by

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ABSTRACT

This report summarizes a Phase I study of automotive-engine induction systems conducted by Battelle-Columbus for the National Air Pollution Control Administration under Contract Number CPA 22-69-9.

The objective of this non-experimental program was to explore incentives and approaches for reducing exhaust emissions by extending the lean-mixture operating limit through improved induction-system performance.

The results of this study indicated that extension of the present limits, made possible by improved air-fuel distribution, can lead to modest but worthwhile reductions in emissions of both unburned hydrocarbons and carbon monoxide and to substantial reductions in the emission of oxides of nitrogen. Minimizing or eliminating the liquid-fuel film on the intake manifold walls was concluded to be the most promising approach to achieving improved distribution. The liquid film can be reduced by employing atomizing devices that will produce droplet sizes approaching 10 to 20 microns under all operating conditions, by designing the induction system for minimum impaction of fuel droplets, and by heating the manifold walls to vaporize the fuel that is unavoidably impacted.

Recommendations include an experimental program comprising the design, fabrication, and evaluation of an induction system with a carburetor employing an alternative, improved atomization device and a low-impaction intake manifold.

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A STUDY OF THE INFLUENCE OF FUEL ATOMIZATION, VAPORIZATION, AND MIXING PROCESSES ON POLLUTANT EMISSIONS FROM MOTOR-VEHICLE POWERPLANTS

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INTRODUCTION

The induction system of a gasoline-fueled, spark-ignited automotive engine, which consists basically of the carburetor and the intake manifold, has as its main functions the metering and mixing of the air and the fuel, and the distribution of this mixture to the individual cylinders. Performance of these functions is complicated by the necessity of accomplishing them over a wide range of engine speeds and vehicle operating modes such as idle, acceleration, steady cruise, deceleration, warm-up, and wide-open throttle.

Ideally, the air-fuel mixture reaching the combustion chambers will be uniform from cycle to cycle and from cylinder to cylinder, and the fuel will be largely vaporized or finely atomized and uniformly mixed in the air stream. Real systems fail to achieve this ideal, as evidenced by observations of fuel flow in transparent carburetors, venturis, and inlet manifolds and by experimental measurements of individual-cylinder air-fuel ratios. This departure from ideal conditions influences power, driveability, fuel economy, and exhaust emissions.

Until recent years, power output, fuel economy, and driveability have been the primary performance objectives in induction-system design and development. Operation with an air-fuel mixture about 8 to 10 percent richer than stoichiometric yields maximum power, while a slightly lean mixture yields best economy. In this mixture range, substantial cylinder-to-cylinder and cycle-to-cycle variations in the air-fuel ratio can be tolerated without serious detrimental effects on engine performance or operation.

Incentives for Induction-System Improvement
to Reduce Exhaust Emissions

Currently, one of the approaches used by automotive-engine manufacturers to reduce exhaust emission levels is the operation of these engines with leaner air-fuel mixtures under idle and cruise (part-throttle) driving conditions. Induction systems have been developed which permit operation with leaner mixtures through closer control of overall air-fuel metering and improved uniformity of air-fuel ratio both between cylinders and from cycle to cycle. However, the extent to which this approach can be used is presently limited by unsteady engine operation which is encountered at still leaner mixtures.

In present engines, individual cylinders approach the lean combustion limit while the average mixture is still satisfactory for good combustion, because of both time variations in mixture air-fuel ratio and cylinder-to-cylinder maldistribution of fuel. Therefore, further improvements in induction-system performance, in terms of more uniform distribution of air and fuel, are necessary in order to extend the lean operating limit. Maldistribution also reduces the beneficial effects of lean-ratio operation on exhaust emissions per se.

Present induction-system designs are based on the use of a venturi-type atomizer, a fuel-metering system that uses the venturi pressure drop to induce and regulate the fuel flow, and a branched manifold to distribute the mixture to individual cylinders. While induction systems have been improved gradually, the basic concepts have remained unchanged for several decades. Further, present designs are the result of experimental development and empirical methods rather than highly analytical approaches.

With the need for further improvement of induction-system performance in response to air-pollution considerations, there is reason to assess the benefits that can be gained by the employment of alternative atomization concepts and a deeper analytical understanding of induction-system phenomena.

OBJECTIVES

This program was conducted for the National Air Pollution Control Administration as a Phase I analytical study. The objectives of this study were (1) to explore the incentives for achieving engine operation with leaner mixtures as a means of reducing exhaust emissions and (2) to develop information on potential means of extending the lean operating limit through improved induction-system design concepts and improved analytical design approaches.

SUMMARY OF STUDY

Scope

The research conducted during this Phase I study was entirely analytical, with no experimental work included. The study was divided into four major subtasks, which were

- (1) Development of mathematical descriptions of induction-system phenomena
- (2) Evaluation of methods for measuring mixture characteristics
- (3) Conception of approaches to the definition and measurement of mixture quality
- (4) Evaluation of alternative atomization methods.

In addition to these four subtasks, a review of the current state of knowledge concerning the influence of induction-system performance on exhaust emissions was made. This latter task, which included evaluation of the effects of air-fuel ratio and mixture nonuniformity on emissions, was aimed at determining the incentive for seeking further improvements in induction systems.

1. Mathematical Study. Mathematical studies included air-flow dynamics, fuel atomization, particle dynamics, vaporization, and liquid-film transport. For each of these system phenomena, existing theories or mathematical expressions were obtained and examined for their applicability to the automotive-engine induction system. In some cases, calculations were correlated with known performance or used to predict potential performance trends.

2. Measurements of Mixture Characteristics. Characteristics which define the state of the air-fuel mixture were identified and measurement techniques were investigated. Both laboratory bench techniques and techniques that might be applied to operating engines were considered. The studies included identification of measurement problems that might be encountered.

3. Definition of Mixture Quality. The task of defining mixture quality was directed toward devising a meaningful way of rating the quality of a mixture that is produced by any fuel-mixture preparation device. A mixture having the ability to be uniformly distributed among all cylinders of a multi-cylinder engine and to achieve complete combustion in the cylinder is a primary objective in induction-system design and development; hence, those mixture properties most closely related to distribution and combustion received major attention.

4. Evaluation of Atomization Methods. The investigation of alternative methods included a review of basic fuel-metering techniques; an evaluation of the design principles, general configurations, and operating characteristics of the principal atomization methods; and an estimation of droplet-size capabilities and energy requirements of these atomization methods. The particular atomization methods selected for detailed evaluation were ultrasonic, spinning-disk, two-fluid (or air atomization), and single-fluid (or pressure atomization). Electrostatic atomization was also evaluated but in less detail because it appears less suitable for the automotive application. General design approaches to apply the alternative atomization methods were devised to illustrate direct replacement of the carburetor.

Emphasis of Study

Technical work was aimed primarily at carbureted induction systems as opposed to fuel-injection systems. The use of fuel injection is recognized as a valid approach to induction-system improvement, and many areas of induction-system technology are common to both types of systems. However, the study of problems unique to the development of fuel-injection systems was not considered to be within the scope of this investigation.

The possibility of improving induction-system performance through innovative design concepts was considered. Some possible innovations in regard to the use of alternative atomization devices are presented in this report; however, principal emphasis was placed on a comprehensive and systematic application of existing technology to the improvement of induction systems, rather than on the generation of innovative design improvements.

Information Sources

During this research program, pertinent information and insight was sought from the published literature, from Battelle experience and reports, and from contacts with industrial organizations and universities. The principal literature sources consulted were: Society of Automotive Engineers publications, Air Pollution Control Association papers, Coordinating Research Council reports, American Petroleum Institute documents and reports, and National Advisory Committee for Aeronautics (now NASA) technical reports and memoranda.

Project personnel visited the following manufacturers: Ford Motor Company, Holley Carburetor Company, American Motors Corporation, Ethyl Corporation, General Motors Corporation, and Chrysler Corporation. In addition, project personnel visited faculty members at The University of Michigan and The Ohio State University. Contacts were also made with engineers from Carter Carburetor Company and the Bendix Corporation, and with faculty members at The University of Wisconsin.

CONCLUSIONS

The results of this Phase I research study lead to the conclusion that the achievement of more uniform and steady air-fuel mixture distribution is essential to minimizing exhaust emissions through engine modifications, and that there are potentially workable approaches to achieving substantial improvements in carbureted induction-system performance.

The following are specific conclusions based on the findings of this study:

- (1) Improved air-fuel distribution that will extend the present lean-mixture-operation limits can potentially lead to modest but worthwhile reductions in emissions of unburned hydrocarbons (HC) and carbon monoxide (CO) and to substantial reductions in the emission of oxides of nitrogen (NO_x). At cruise conditions, operation with a uniform air-fuel ratio in the range between 18:1 and 20:1 appears optimum for minimizing HC and CO emissions. Operation at 20:1 might reduce NO_x emissions by as much as 80 percent from present levels.
- (2) The most promising approach to achieving improved distribution that was identified in this study is through minimizing or eliminating the presence of a liquid-fuel film on the intake manifold walls. The flow of this liquid film is inherently unsteady, and the resulting stratification of the air-fuel mixture gives rise to geometric maldistribution. The liquid film can be reduced by the following:
 - Use of atomizing devices that will produce droplet sizes approaching 10 to 20 microns under all operating conditions. Air atomizers and ultrasonic atomizers appear most promising and potentially feasible; they can be used in conjunction with conventional fuel-metering concepts.

Conventional venturi-type atomizers are potentially capable of producing the desired levels of atomization only at the high venturi air velocities obtained at high engine power levels.

- Design of induction systems for minimum impaction of fuel droplets. This can be accomplished by (a) designing for minimum turning angle of the air-fuel mixtures, (b) designing for moderate air velocities, (c) designing manifold flow-passage cross-sectional geometry for minimum impaction, and (d) the use of low-impaction alternatives to the conventional butterfly-type throttle plate.
- Heating of manifold walls to vaporize fuel that is unavoidably impacted.

Further work involving an experimental program will be required to determine the most feasible combination of atomization device, droplet-size characteristics, manifold impaction characteristics, and heat-addition scheme.

- (3) Existing mathematical analyses for predicting droplet impaction are based on the assumption of idealized one-dimensional steady air flow. Experimental studies are needed to support the development of mathematical theory which includes the effects of unsteady flow and two- or three-dimensional air-velocity profiles.
- (4) There does not appear to be a strong incentive at present to develop a mixture-quality rating scheme. A corollary of the second conclusion presented above is that the air-fuel mixture properties of principal importance are
 - The percentage of the fuel in droplet sizes small enough to remain suspended in the air while traversing the intake manifold
 - The percentage of the fuel in vapor form
 - The homogeneity of the mixture.

These properties can be measured individually, and their relationship to induction-system performance is generally understood. If, in the future, a convenient means for rating the quality of mixture produced by carburetors appears desirable, a simple apparatus based on an impaction pattern on a target in a plenum could be used.

RECOMMENDED PROGRAM

The extension of this investigation into a Phase II experimental program is recommended to demonstrate the technical feasibility of improved induction-system design for reducing exhaust emissions. This program would consist of (a) experimental studies leading to the design and construction of an induction system incorporating an improved atomization device and a low-impaction intake manifold and (b) a laboratory evaluation of the effectiveness of the system for improving fuel distribution to extend the lean operating limit and to reduce exhaust emissions. The program would proceed as follows:

- (1) Droplet Transport. Bench studies of droplet-transport characteristics would be the first task. This would include experimental verification of analytical predictions of impaction characteristics based on steady-flow theory, investigation of the effects of unsteady air flow and droplet-size dispersion on droplet dynamics, and investigation of the effects of manifold geometry on impaction.

Single-duct test sections of various cross-sectional areas, geometries, bend radii, and turning angles would be aspirated by a reciprocating compressor or other device capable of inducing a controlled unsteady air flow through a laboratory fuel-atomization device for producing fuel droplets of controlled size.

The objective of this part of the study would be to develop information on which to base the selection of an optimum mean droplet size and a favorable intake-manifold configuration.

- (2) Mixture Measurements. In connection with both the activity described above and subsequent tasks of the program, it would be necessary to construct a laboratory apparatus for measuring the significant mixture properties, namely, droplet size, degree of vaporization, and stratification. This would be accomplished using instrumentation techniques described in this report.
- (3) Atomizing Carburetor. The next step would be the design, preliminary development and demonstration of a "carburetor" having improved atomization characteristics. The type of atomizer selected should be the simplest and most practical system capable of delivering the desired optimum droplet size.

In terms of simplicity and compatibility with conventional fuel-metering systems, the piezoelectric ultrasonic atomizer appears most desirable, followed by impinging-air-jet and Hartmann-whistle devices. However, their minimum-droplet-size capabilities are in the opposite order. On the basis of present knowledge, it appears that the piezoelectric ultrasonic atomizer may have limited flow capacity at the probable desired droplet size. In this case, impinging-jet atomization would be the most likely selection.

It may be possible to adapt the fuel-metering section of a commercially available carburetor in the construction of either of these devices. Since proof of principle is the desired objective, it may not be necessary for the test device to incorporate all the features and functions required of a production unit, such as idle and maximum-power-enrichment systems. The resulting device would be checked for performance on a test stand and, if necessary, modifications would be made to achieve the desired atomization characteristics.

- (4) Experimental Intake Manifold. Concurrently, a low-impaction intake manifold would be designed and fabricated. It would be desirable for this manifold to fit a standard V-8 engine within the same space allocated to the conventional induction system. Windows and ports would be designed into the manifold for observation and instrumentation.
- (5) Engine Distribution and Emissions. The final step of the program would be a dynamometer-test-cell evaluation of the system on a standard production engine. Basic tests would include measurements of exhaust emissions and cylinder-to-cylinder distribution under various cruise conditions with progressively leaner mixtures, with both the standard and experimental induction systems, until a lean operating limit was reached.

Additional tests would include measurement of fuel distribution and exhaust emissions at various speeds with wide-open throttle at a single maximum-power mixture ratio.

INDUCTION-SYSTEM PERFORMANCE AND EFFECT ON EMISSIONS

- **Effect of Air-Fuel Ratio on Exhaust Emissions**
- **Studies of Air-Fuel Mixture Distribution**
 - **Time Variations**
 - **Geometric Maldistribution**
- **Effect of Maldistribution on Exhaust Emissions**
- **In-Cylinder Nonhomogeneity**

INDUCTION-SYSTEM PERFORMANCE AND EFFECT ON EMISSIONS

Effect of Air-Fuel Ratio on Exhaust Emissions

Extensive experimental studies have been conducted over the past several years to determine the relationship between air-fuel ratio and exhaust emissions. These studies have been conducted on single-cylinder research engines and on multicylinder automotive engines, with tests on the latter conducted both in the laboratory and on the road. The results have been remarkably similar in spite of the numerous differences in engines and operating conditions. In this study, the published data from a number of previous studies^{(1-8)*} were evaluated and summarized to yield a representative picture of the principal automotive exhaust emissions as influenced by air-fuel ratio.

Figure 1 summarizes the results of this evaluation, presented in terms of nondimensionalized curves of CO, HC, and NO_x as a function of air-fuel ratio. These curves are drawn as bands in recognition of variations in the data. The data of Figure 1 represent conventional multicylinder engines without exhaust emission controls (pre-1968) and were obtained while operating at part-throttle cruise conditions between 20 and 60 mph. The nondimensionalized presentation of the data permits general trends to be observed readily without concern for specific emission levels, which tend to be different for different engines and which have been changing from year to year as exhaust controls have been applied.

In general, CO emissions show a decreasing trend with increasing air-fuel ratio, leveling off to a minimum value at about 18 to 1^{**} air-fuel ratio and beyond. HC emissions also are seen to decrease with increasing air-fuel ratio. However, HC emissions reach a minimum around 18 or 19 to 1 air-fuel

* See References and Bibliography.

** There appears to be no standard convention in the automotive industry for expressing the fuel concentration in the air-fuel mixture. Air-Fuel ratio, fuel-air ratio, air-fuel equivalence ratio, fuel-air equivalence ratio, and percent excess air are all used. Air-fuel ratio, expressed as the mass of dry air per unit mass of fuel in the mixture, was selected for use in this report.

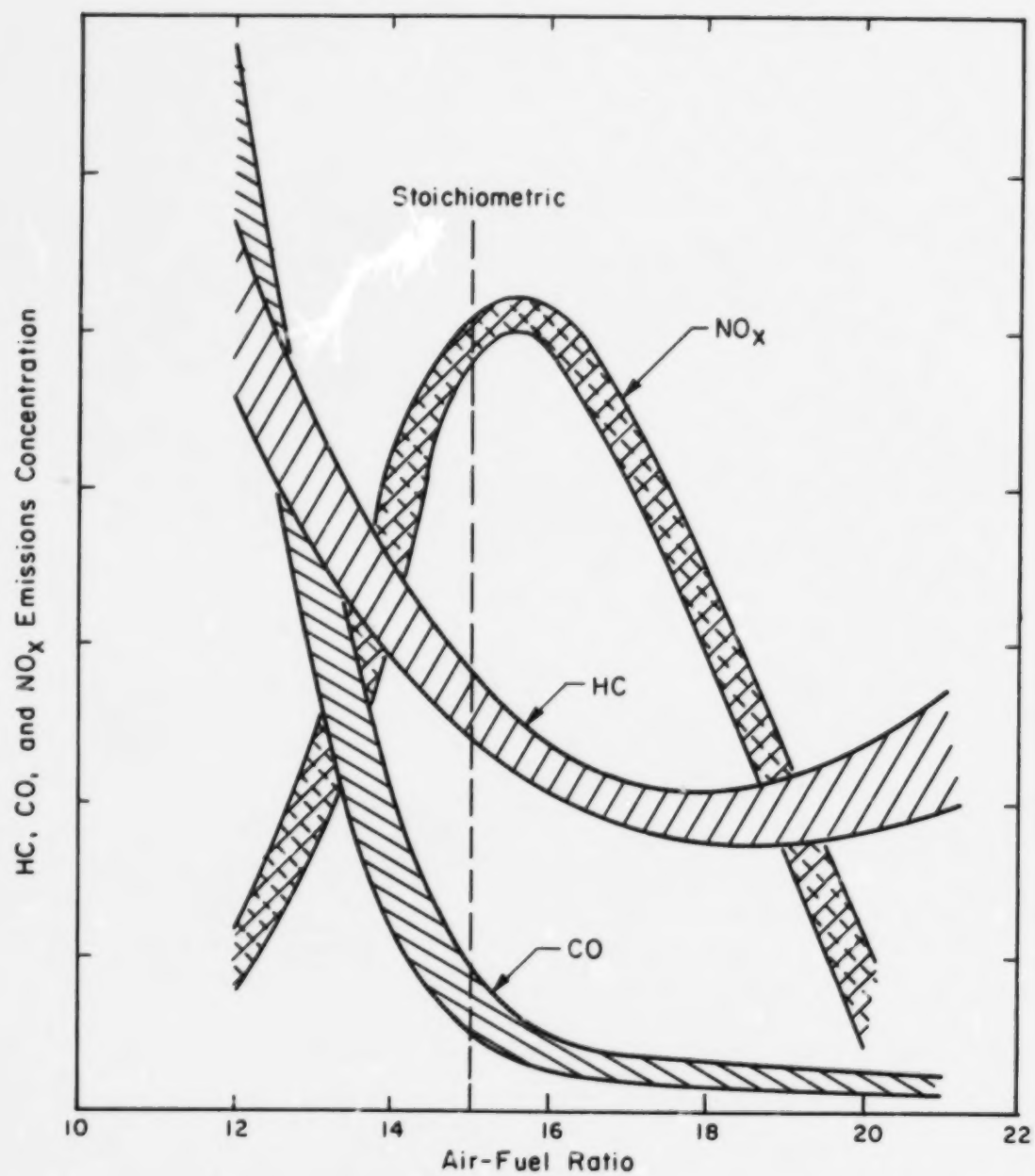


FIGURE 1. EFFECT OF AIR-FUEL RATIO ON EXHAUST EMISSIONS AT ROAD-LOAD OPERATION

ratio and then begin to increase again. This phenomenon of increasing HC at higher air-fuel ratios is due to operating too close to the lean limit of combustion and is made worse by maldistribution.

Perfect cylinder-to-cylinder distribution would extend the point of minimum HC emissions to a higher air-fuel-ratio value. NO_x emissions increase to a maximum value as the air-fuel ratio is increased to about 15 or 16 to 1 and decrease at higher air-fuel ratios. NO_x emissions are known to be related to peak combustion temperatures, which are lower with lean mixtures.

Carburetor and intake-manifold development over the past several years has led to the current practice of operating with an air-fuel ratio of 16 or 17 to 1 at part throttle--with at least one production engine operating as high as 18 to 1. The move to leaner operation would appear, from the emission curves of Figure 1, to have played a significant role in achieving the present levels of HC and CO emissions. Part-throttle operation at even leaner air-fuel ratios, say as high as 20 to 1, would result in small additional decreases in HC and CO emissions, assuming good distribution. However, NO_x emissions could be reduced substantially by leaner operation. The reduction in NO_x emissions at 20 to 1 air-fuel ratio might be as much as 80 percent from present levels.

Although the foregoing data and discussions are based on part-throttle operation, there is evidence to indicate that exhaust-emission trends are similar at idle and at wide-open-throttle conditions. The data available on idle operation indicate that both HC and CO emissions reach minimum values at lower air-fuel ratios than at part-throttle cruise operation, and that the HC emissions have a greater tendency to increase as air-fuel ratio is increased beyond the minimum point. At wide-open throttle the minimum HC emissions occur at about the same air-fuel ratio as at part throttle, but there is less increase at higher air-fuel ratios. CO emissions are relatively unchanged at wide-open-throttle operation.

NO_x emissions at wide-open throttle are higher than at part throttle, but the peak value occurs at about the same air-fuel ratio. NO_x emissions do not appear to be a problem at idle operation. Studies have indicated that

NO_x is very low during idle, perhaps less than 10 percent of the peak value at road load, and remains low over the range of air-fuel ratios acceptable for idle operation.⁽⁹⁾

Studies of Air-Fuel Mixture Distribution

Mixture nonuniformity at the individual-cylinder inlet valve occurs as a result of one or more of the following phenomena: (1) time variations in air-fuel-ratio delivery, (2) geometric maldistribution of the fuel and air between cylinders, and (3) nonhomogeneity in the mixture. Of these three phenomena, geometric maldistribution is the easiest to detect and has received the greatest attention. However, all three are detrimental to performance to some degree and have been of concern to induction-system designers.

Time Variations

Fuel travels in the intake manifold in the form of vapor, entrained liquid droplets, and a liquid film on the manifold walls. Time variations in air-fuel ratio delivery are caused primarily by inherent instabilities in the flow of the liquid-fuel film. There is also some evidence that the carburetor itself delivers fuel at an unsteady rate under some conditions, particularly at low air velocities through the venturi.

An undesirable consequence of time variation of the air-fuel ratio is power surge. Unsteady air-fuel ratio delivery will result in torque pulsations which, if great enough, will give rise to a jerky feeling to the occupants of the vehicle.

In a fuel distribution study conducted by Yu⁽¹⁰⁾, a gas-sampling valve was developed and used to obtain individual-cycle gas samples directly from cylinders of a V-8 engine. The samples were analyzed by gas chromatography to determine air-fuel ratio. The time variation of air-fuel ratios measured by this technique at part-throttle operation ranged from 16.8 to 19.5, for a

spread of 2.7 ratio numbers (15 percent based on the measured average air-fuel ratio) using iso-octane fuel. The spread increased slightly at higher average air-fuel ratios, increased significantly at higher manifold vacuums, and decreased significantly with increased fuel volatility.

Geometric Maldistribution

Geometric maldistribution, or cylinder-to-cylinder mixture variations, results from the flow of a stratified air-fuel mixture through asymmetrical or branched carburetor and inlet-manifold configurations. Geometric maldistribution includes not only air-fuel ratio variations but also, as a consequence of the partial vaporization of the fuel, a maldistribution of fuel chemical species and additives. This can result in some cylinders receiving mixtures lower in octane rating than other cylinders receive.

Geometric maldistribution can result in loss of power and efficiency if some cylinders are caused to operate with an over-rich mixture. Geometric maldistribution may also contribute to engine roughness as a result of an imbalance of power pulses between cylinders, particularly under part-load operating conditions. Furthermore, it is possible for knock or misfiring to occur in some cylinders if they receive insufficient antiknock additive or too lean a mixture.

In the study by Yu previously mentioned⁽¹⁰⁾, data on time-averaged air-fuel ratio were obtained from each cylinder of the engine by exhaust-gas analysis. The air-fuel ratio spread due to geometric maldistribution was 2.4 ratio numbers when the measured average air-fuel ratio was 17.7. Complete vaporization of the fuel (simulated by using propane), coupled with induced swirl of the injected fuel in the carburetor venturi and an additional mixing volume between the carburetor and the manifold, reduced the air-fuel ratio spread to 0.3 ratios, with a measured average air-fuel ratio of 18.4.

In a study of fuel distribution in a 1956 V-8 truck engine conducted by Cooper, et al⁽¹¹⁾, radioactive tracers were used to "tag" individual fuel components. The distribution of these "tagged" components, including certain

additives such as tetraethyl lead (TEL), to individual cylinders was determined by measuring the radioactivity in the exhaust of each cylinder. The study included the effect of various fuel factors such as volatility and heat of vaporization, operating factors such as engine speed and acceleration, and engine-design factors such as throttle-plate position and carburetor-venturi configuration.

Figure 2 illustrates the maldistribution of air-fuel ratios and gasoline additives measured in the Cooper study for one particular engine operating condition. The maldistribution of TEL follows quite closely the maldistribution of air-fuel ratios. Cylinders rich in fuel (the numerically lower air-fuel ratios) are also rich in TEL, which indicates that only moderate vaporization of the fuel is taking place in the induction system. The TEL, having a boiling point near the top of the distillation range for the gasoline, will follow the liquid fuel.

There are actually two scavengers represented in the curve of Figure 2, ethylene dibromide and ethylene dichloride, with boiling points above and below the midrange of the fuel. Their combined distribution is fairly uniform.

The results of the Cooper study also indicated that maldistribution of air-fuel ratios was increased by increasing the average air-fuel ratios; by increasing the fuel latent heat and specific gravity; by decreasing fuel volatility; and by decreasing the engine speed. Throttle position appeared to have the greatest effect on maldistribution. With the throttle plate half open, as was the case for the data of Figure 2, the cylinder-to-cylinder air-fuel ratios ranged from 9.8 to 17.9, with an average of 12.9. The throttle-plate orientation, with its shaft perpendicular to the longitudinal axis of the engine, tended to deflect the fuel droplets toward the front cylinders, causing them to run rich. When the throttle-plate shafts were relocated to be parallel with the engine crankshaft, the air-fuel ratio spread was reduced to only about 1/4 the previous value at the half-open throttle position.

A study was conducted by Jones and Gagliardi⁽¹²⁾ to determine the effect of complete vaporization and mixing of the fuel and air on cylinder-to-

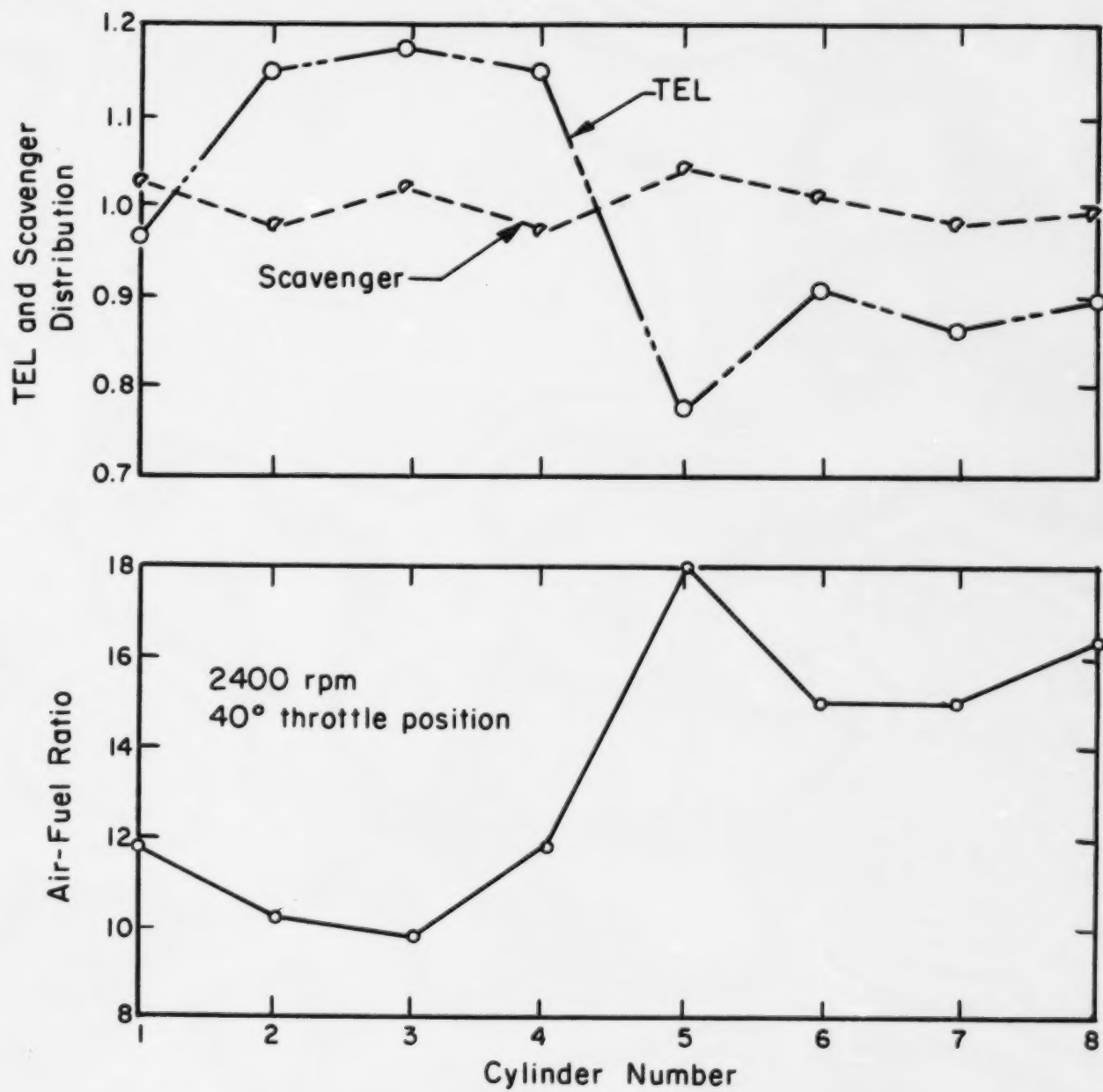


FIGURE 2. GEOMETRIC DISTRIBUTION OF AIR-FUEL RATIO, TEL, AND LEAD SCAVENGERS IN AN 8-CYLINDER ENGINE

cylinder air-fuel ratio distribution. Tests were run with 6- and 8-cylinder engines using the Federal 7-mode cycle. The standard carburetors were used in one set of tests, and an external mixing and vaporizing tank was used in another set of tests. Individual-cylinder air-fuel ratios were determined by exhaust-gas analysis.

Figure 3 shows the cylinder-to-cylinder air-fuel ratio variations for both carburetor operation and vaporizing-tank operation, with the 8-cylinder engine operating at the 30-mph cruise mode. Maldistribution is almost completely eliminated when the fuel is completely vaporized and completely mixed before entering the inlet manifold. The maximum air-fuel ratio spread with carburetor operation is 2.3. At other vehicle operating modes the air-fuel ratio spread varied from 1.3 to 2.0 with the carburetor, and from 0.3 to 0.7 with vaporizing-tank operation.

Effect of Maldistribution on Exhaust Emissions

In a study by Robison and Brehob⁽¹³⁾, the effects of individual-cylinder mixture quality on exhaust emissions were determined by a similar approach using a vaporizing tank. This was accomplished by comparing the cylinder-to-cylinder distribution of the mixture and of the HC and CO exhaust emissions for the normal carburetor-induced fuel and for the vaporized fuel, using a standard 6-cylinder automotive engine. The vaporized fuel was premixed with the air before entering the induction system to insure good distribution. Among the important conclusions of this study was the observation that significant reductions in both HC and CO emissions were obtained with the vaporized fuel.

Figure 4 illustrates the relationship between emissions and air-fuel ratio at part-throttle operation as reported by Robison and Brehob. Reductions on the order of 10 to 20 percent were demonstrated for both HC and CO emissions.

Figure 5 shows similar data for wide-open-throttle (WOT) operation. In both cases, CO emissions do not appear to be affected by mixture quality at average air-fuel ratios greater than 16.

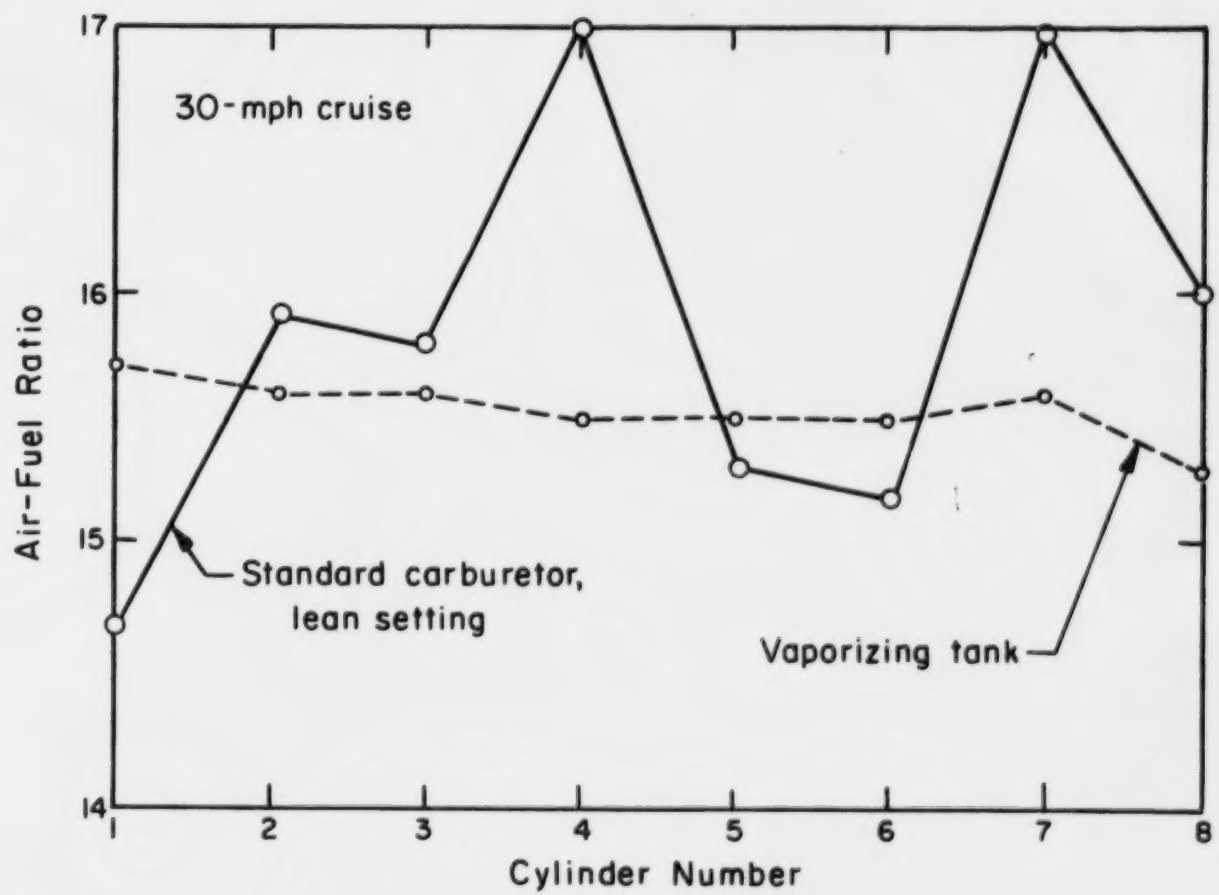


FIGURE 3. GEOMETRIC DISTRIBUTION OF AIR-FUEL RATIO FOR CARBURETED AND FOR VAPORIZED FUEL IN AN 8-CYLINDER ENGINE

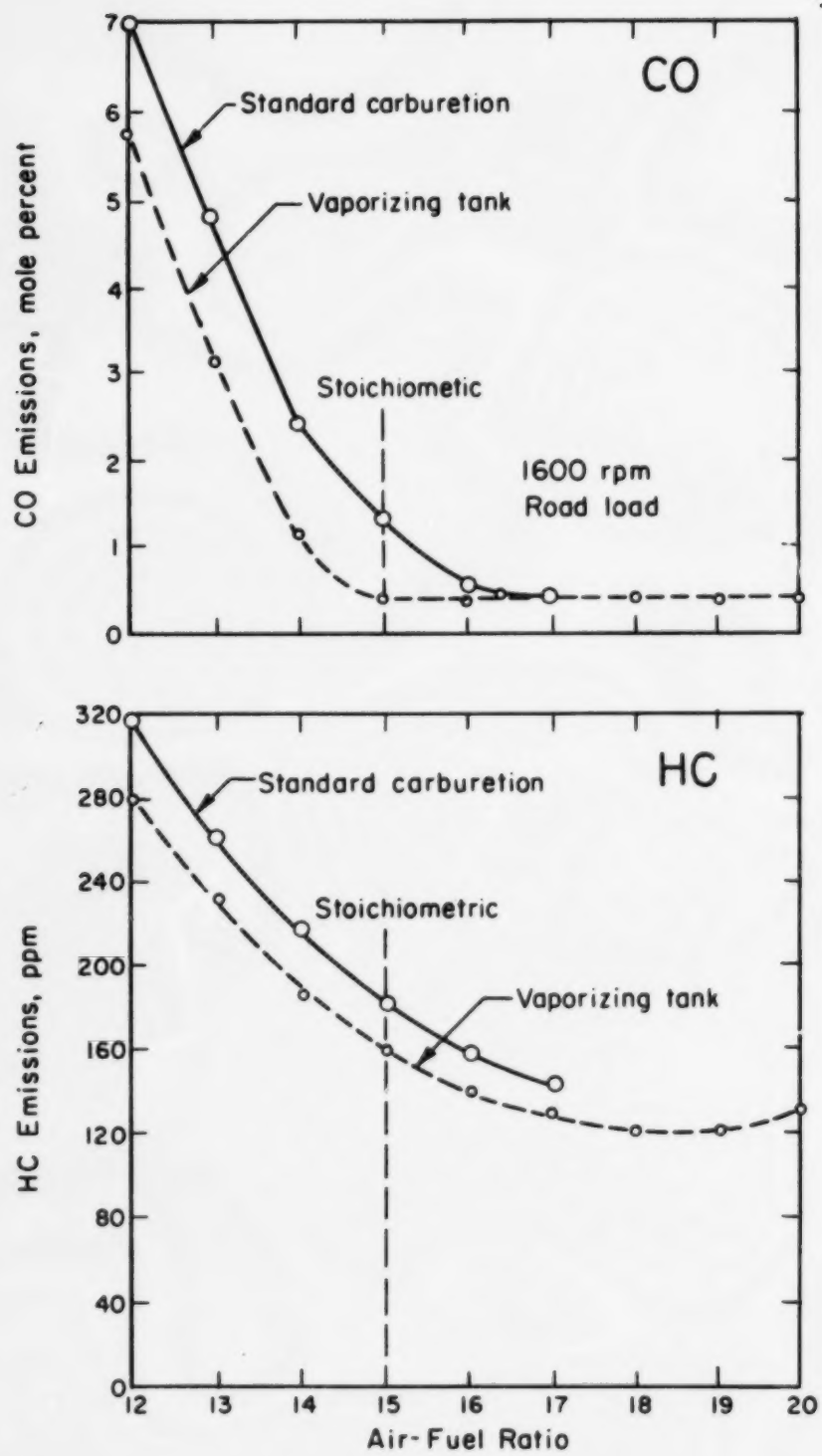


FIGURE 4. EFFECT OF IMPROVED GEOMETRIC DISTRIBUTION ON HC AND CO EMISSIONS AT ROAD LOAD

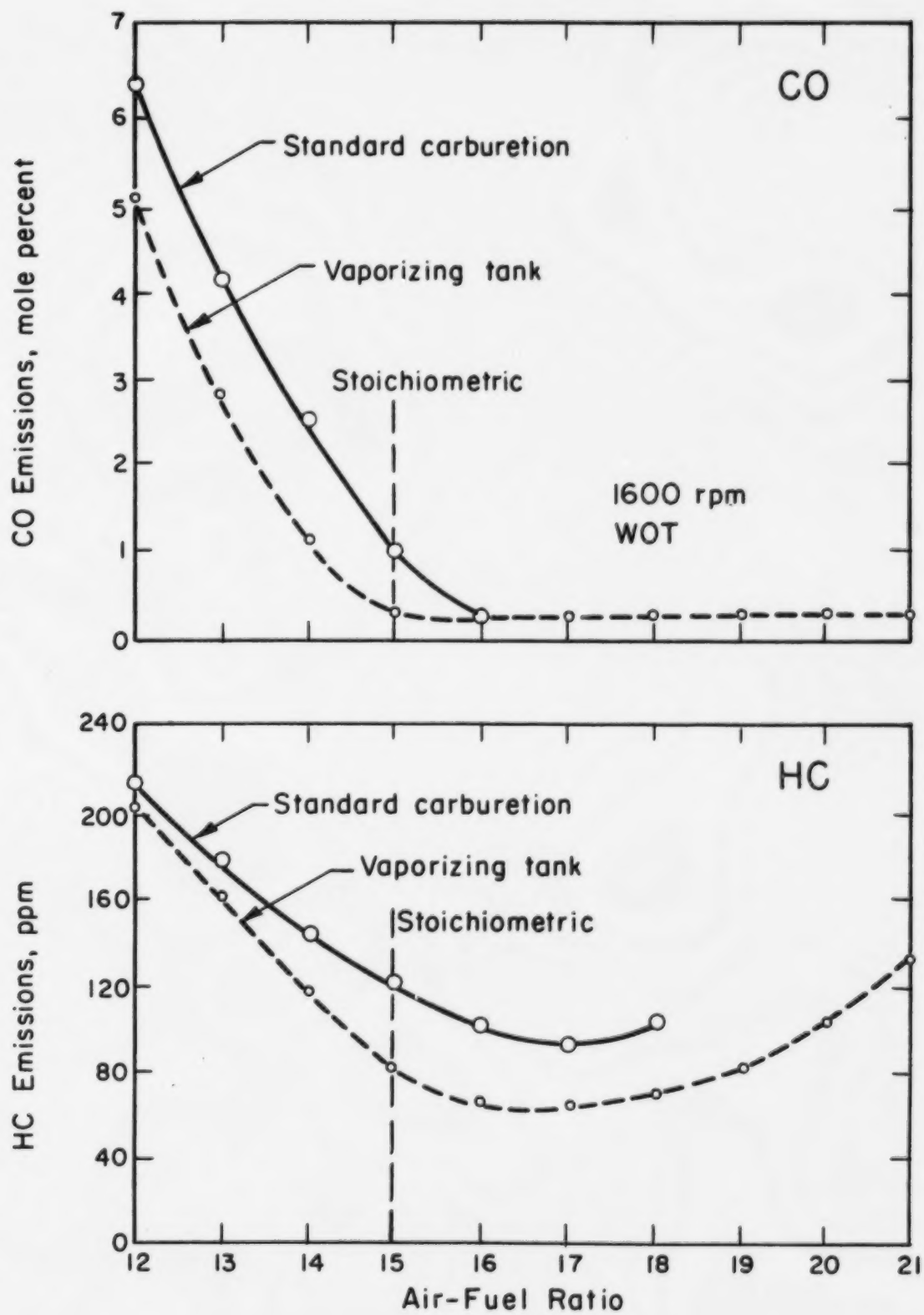


FIGURE 5. EFFECT OF IMPROVED GEOMETRIC DISTRIBUTION ON HC AND CO EMISSIONS AT WOT

Figure 6 shows cylinder-to-cylinder air-fuel-ratio distribution for the same tests represented by the data of Figures 4 and 5. Geometric maldistribution is poor at wide-open throttle with standard carburetion, but fairly good at part throttle. Distribution with vaporized and premixed fuel is good at all operating conditions. At the worst distribution, occurring at 1600 rpm WOT with standard carburetion, the cylinder-to-cylinder air-fuel ratio varied from 10.1 to 16.8 with an average of about 13.1. For that same operating condition, the cylinder-to-cylinder HC emissions varied from about 480 to 990 ppm and the cylinder-to-cylinder CO emissions varied from about 0.3 to 9.0 mole percent.

An important conclusion of the Robison and Brehob paper relates to the lean limit for misfire-free operation --misfire results from excessively lean mixtures, among other things. If cylinder-to-cylinder distribution is poor, the tendency to misfire at lean average air-fuel ratios will be greater than with good distribution. Misfiring will cause exhaust emissions to increase. Vaporized-fuel operation raised the misfire lean-limit as much as 4 air-fuel ratio numbers at medium to high-speed road-load operating conditions.

In-Cylinder Nonhomogeneity

Nonhomogeneity in the mixture arriving at the inlet valve results from failure of the fuel vapor and fuel droplets to mix uniformly with the air. Any fuel on the wall would result in nonhomogeneity. A nonuniform mixture entering the cylinder is believed to contribute to variations in flame speed which can lead to cylinder-pressure fluctuations and, ultimately, to increased emissions. However, the relationship between in-cylinder nonhomogeneity and combustion phenomena is not well understood.

In summary, there is considerable experimental evidence that mixture nonuniformity between cylinders increases HC and CO emissions, especially around stoichiometric air-fuel ratio. Of equal or greater importance is the limitation imposed by maldistribution on lean-mixture operation. Operation at

air-fuel ratios of 20 or greater can result in significant overall exhaust emissions reduction, but this can be accomplished only if fuel and air distribution are greatly improved, and time-variation of air-fuel ratio is minimized.

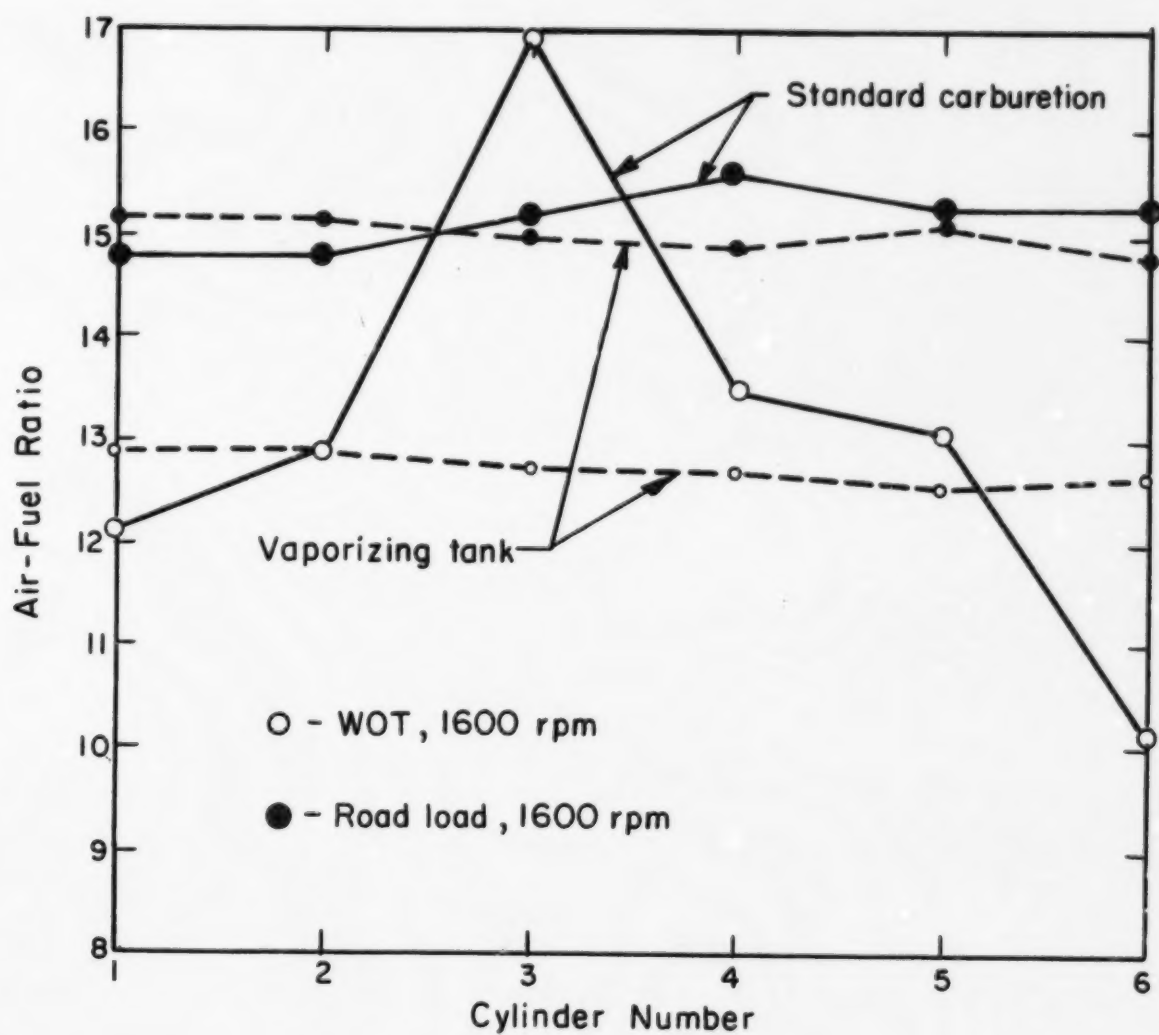


FIGURE 6. GEOMETRIC DISTRIBUTION OF AIR-FUEL RATIO FOR CARBURETED AND FOR VAPORIZED FUEL IN A 6-CYLINDER ENGINE

MATHEMATICAL APPROACHES TO INTAKE MANIFOLD DESIGN

- Induction-System Phenomena
 - Induction-System Geometry
 - Air-Flow Phenomena
 - Fuel Flow

- Air-Flow Mathematical Models
 - Plenum Model
 - Straight-Pipe Model
 - Helmholtz-Resonator Model
 - Organ-Pipe Model
 - Composite Model

- Fuel-Droplet Dynamics
 - Droplet Deposition Mechanisms
 - Inertial Impaction of Droplets
 - Droplet Impaction in a Conventional Induction System

- Vaporization and Heat Transfer
 - Equilibrium Vaporization
 - Droplet Vaporization
 - Manifold Wall-to-Air Heat Exchange
 - Vaporization From Surface Films

- Liquid-Film Transport

- Incentives for Further Analytical Studies of Air and Fuel Transport

MATHEMATICAL APPROACHES TO INTAKE MANIFOLD DESIGN

Induction-System Phenomena

A phenomenological understanding of the processes that occur in the induction system is a prerequisite for a mathematical approach to induction-system design. The present state of knowledge of the phenomena appears to have been gained by a combination of that which is intuitively obvious, visually observed, can be deduced from calculations, and observed results of induction-system performance such as volumetric efficiency and air-fuel distribution. It appears that little work has been done toward actually measuring local and instantaneous pressures, flow rates, fuel droplet sizes, percent of fuel vaporized, liquid-fuel film thickness, holdup, or slip ratio.

Induction-System Geometry

Figures 7, 8, and 9 show a typical configuration of a modern V-8 engine induction system in simplified form. Figure 7 shows a schematic top view of the most widely used manifold geometry. Two separate "H"-shaped groups of passages are used, each communicating with one barrel (or pair of barrels) of the carburetor and with 4 cylinders. The two halves of the manifold are placed at different levels, as shown in Figure 8, which is a composite section view showing one branch of each half. As pictured, the intake ports in the cylinder heads are, in effect, extensions of the manifold.

A rudimentary representation of a conventional carburetor is shown in Figure 9. Fuel is introduced and "atomized" at the throat of a small venturi, and, in turn exits into the throat of a second and larger venturi--sometimes three are used. The air-pressure depression produced by the venturis provides the motivating force for the fuel flow and also provides a measure of the mass flow rate of air. An air bleed is usually introduced into the fuel passage between the carburetor float bowl and the fuel nozzle, so the fuel is delivered as alternating slugs of air and fuel. An orifice in this passage provides the basic fuel-metering function.

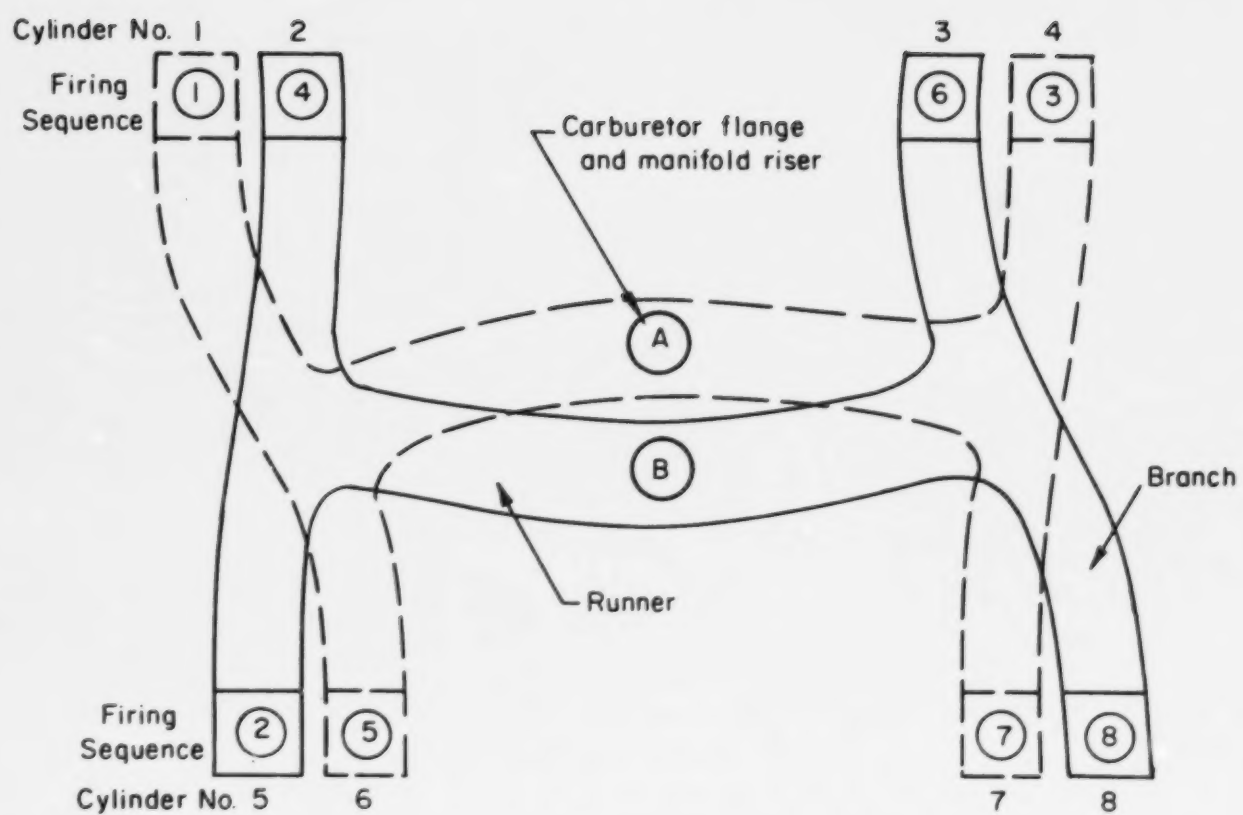


FIGURE 7. SCHEMATIC PLAN VIEW OF TYPICAL V-8 MANIFOLD

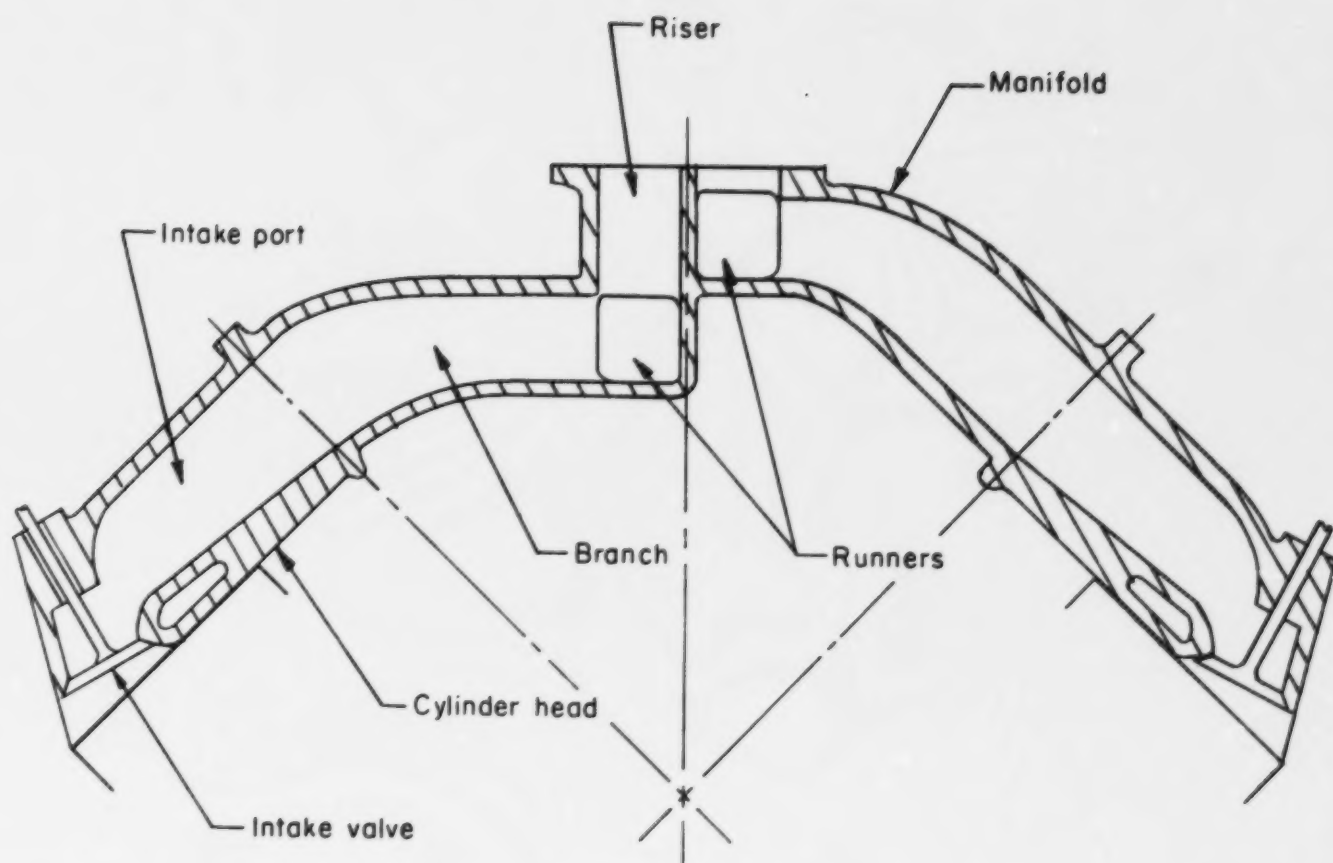


FIGURE 8. TRANSVERSE SECTION OF TYPICAL V-8 MANIFOLD

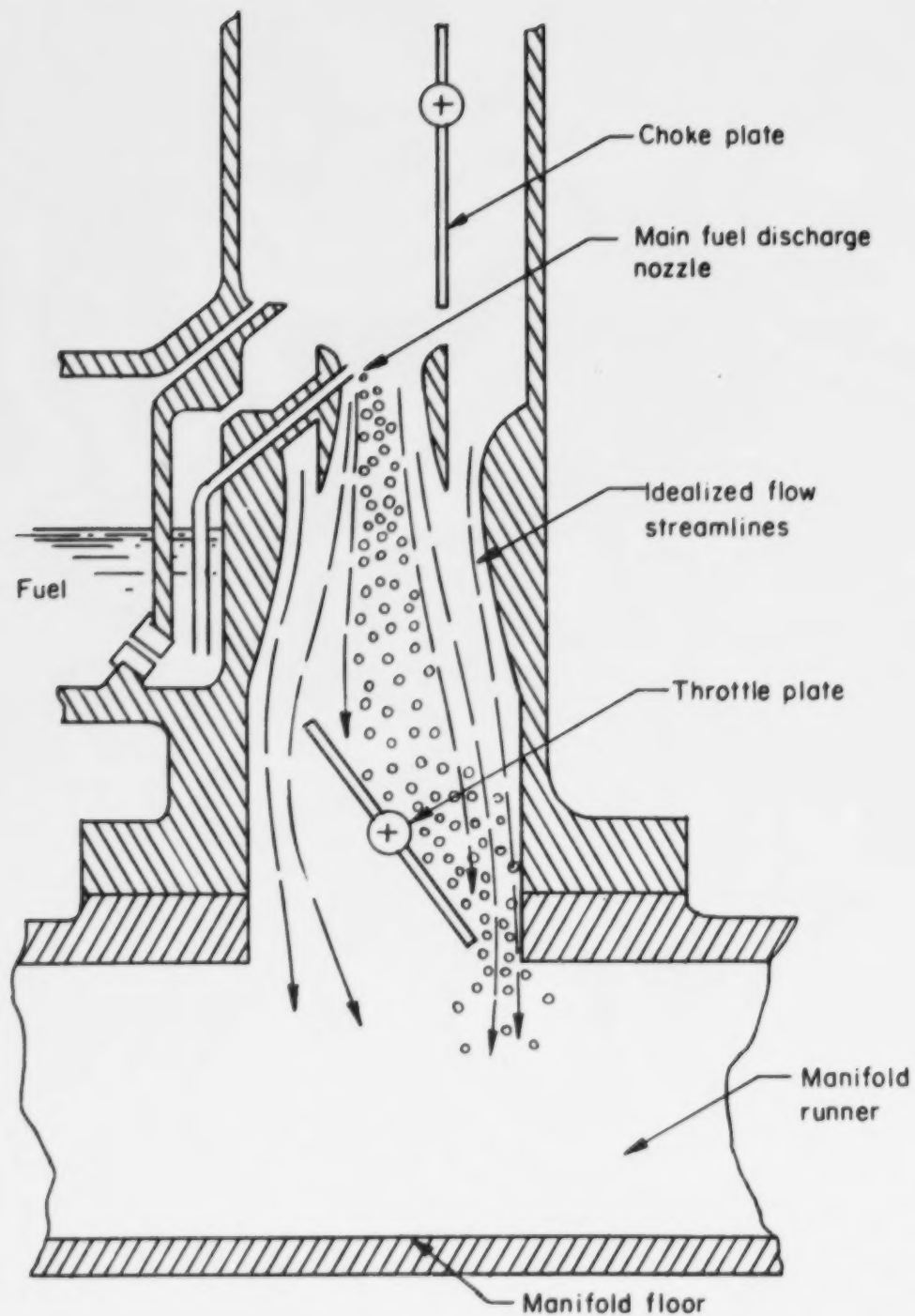


FIGURE 9. SCHEMATIC REPRESENTATION OF FLOW PASSAGES THROUGH A CONVENTIONAL AUTOMOTIVE CARBURETOR

The choke is wide open during operation with the engine warmed up. For cold starting, the partially closed choke induces an additional air-pressure depression in the venturi section which enriches the air-fuel mixture. The function of the throttle plate is to govern the mixture pressure inside the intake manifold.

Air-Flow Phenomena

With an air-fuel ratio on the order of 15 to 1, the air flow is not strongly affected by the presence of the fuel. Therefore, it is possible for practical purposes to ignore the presence of the fuel and consider the flow of air separately.

Flow Fluctuations. In a conventional V-8 induction system, each half of the manifold serves alternate cylinders in the firing sequence; thus, the intake pulses are evenly spaced for each half. With a two-revolution cycle, a half-revolution intake stroke, and 4 cylinders per half manifold, the flow out of the manifold will take place as shown in Figure 10. Ideally, the outflow momentarily goes to zero between intake strokes. (Outflow pulses in a 6-cylinder-engine manifold will overlap.) It is interesting to note that, with the firing sequence indicated in Figure 7, flow pulses are evenly spaced in the runners of the "A" half of the manifold, but not in the "B" half. The "B" half gets two consecutive pulses in one runner, then two in the other runner. It appears that all conventional V-8s have this characteristic.

Flow at the carburetor venturi will fluctuate as a consequence of the pulsed outflow from the manifold. At high intake vacuum, the flow will be continuously inward at the throttle and the flow fluctuations will be relatively small. When the outflow to the cylinder which is going through the intake stroke is greater than the flow through the throttle, the cylinder will draw mixture from the rest of the intake manifold. During the portion of the intake stroke when the flow into the cylinder is lower than the flow through the throttle, mixture will flow back into the rest of the manifold.

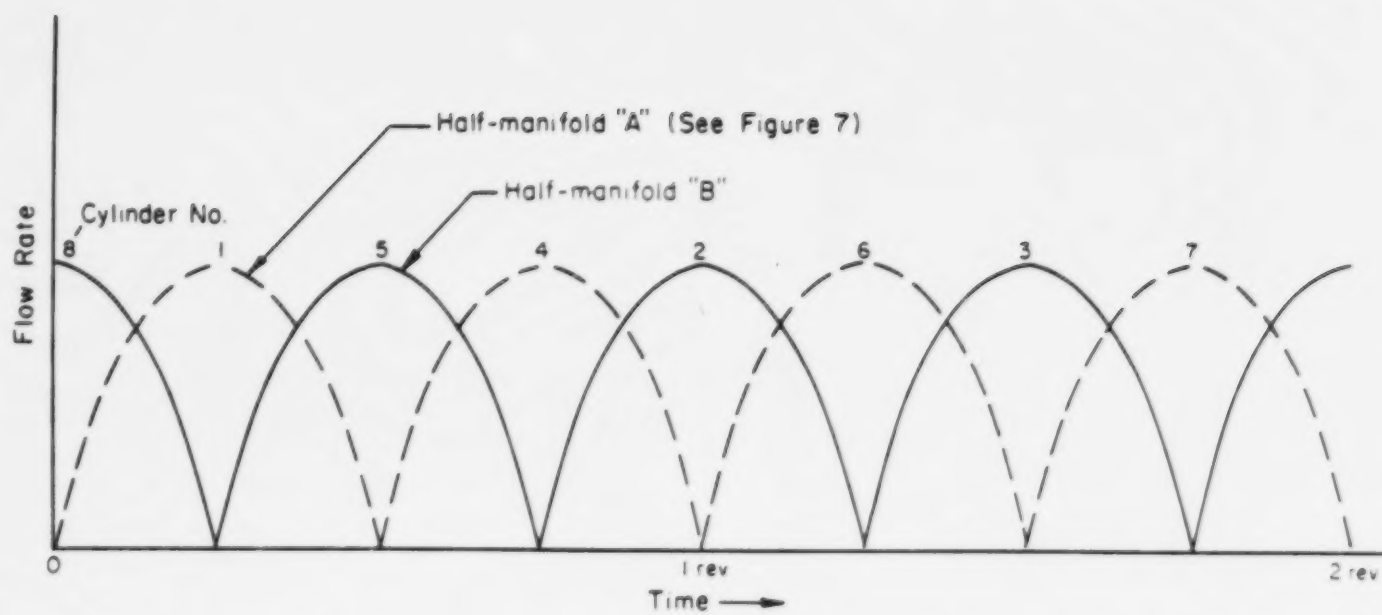


FIGURE 10. IDEALIZED V-8 MANIFOLD OUTFLOW CHARACTERISTICS

At wide-open throttle (WOT), with minimum flow restriction at the throttle plate, flow pulsations at the carburetor will be much more pronounced, and ideally drop to zero between pulses, and backflow may even occur due to dynamic forces on the air in the manifold.

Dynamic Effects. Conventional manifolds are not "tuned"; that is, there is usually no attempt made to obtain resonant "organ-pipe" phenomena to affect cylinder charging. This phenomena is sometimes utilized in high-performance or racing engines, but has the disadvantage of being effective only over a narrow speed range. However, dynamic effects are important in conventional manifold design. When the intake valve of a particular cylinder opens and air begins to flow into the cylinder, the column of air between the carburetor and the cylinder undergoes an acceleration. This causes the air pressure at the valve to drop below the pressure at the throttle bore. After the flow at the valve passes its peak and begins to decrease, deceleration of the column of air causes an increase in pressure at the valve; this has a beneficial effect on volumetric efficiency. Since the intake valve closes well beyond bottom dead center (BDC) with conventional valve timing, there will usually be some back flow through the intake valve at low engine speed.

The abrupt closing of the valve and sudden deceleration of the air column will shock the air into a weak, highly damped "organ-pipe" oscillation. Although this is a resonant phenomena, the frequency is usually several times the fundamental frequency of the intake process, and any effects on cylinder charging are extraneous. The amplitude of the manifold-pressure variations induced by volumetric, inertia, and "organ-pipe" effects are each on the order of up to a few psi.

Because of these dynamic effects, as well as because of differences in flow friction, cylinder-to-cylinder distribution of air is normally significantly unequal.

Velocity Profiles. The air flow is, of course, three dimensional. In six-cylinder manifolds, the air has been observed to have a strong swirl component induced by the throttle. In V-8 manifolds, the throttle plate is

usually mounted at right angles to the runner. This causes asymmetry in the flow, but no gross swirl. Many studies of secondary flows induced in bends have been conducted, and the same kind of effects can be expected to occur in intake manifolds. The pulsating nature of the flow probably tends to inhibit the growth of the boundary layer at the manifold wall.

Typical Velocities. To characterize manifold air flow, we may assume that the following parameters are representative of a modern V-8 engine:

Displacement	340 cu in.
Manifold branch flow area	2.2 in. ²
Primary throttle bore diameter	1.44 in.
Secondary throttle bore diameter	1.69 in.
Maximum speed	5000 rpm
Idle speed	650 rpm
Typical flow-path distance between throttle bore and intake valve	12 in.
Average intake-passage flow area	2.5 in. ²

Assuming 120 F air and volumetric efficiency slightly in excess of 0.80, the following flow parameters result:

	<u>Max Speed</u>	<u>Idle</u>
Peak air velocity in manifold branch, ft/sec	340	44
Peak Reynolds No. in manifold branch	370,000	48,000
Average air velocity at throttle bore, ft/sec	124	38

At 5,000 rpm, the individual-cylinder intake-process duration is 6 milliseconds; at 650-rpm idle it is 46 milliseconds.

The effective swept volume per cylinder is about 35 cu in.; the volume of one direct flow path from throttle bore to intake valve is about 30 cu in. Consequently, a generalization that can be made is that the volume of air-fuel mixture pulled into the cylinder during one individual-cylinder intake stroke

is about the same as the volume of the manifold from the throttle plate to the intake valve. One stroke loads the manifold, the next one pulls the charge into the cylinder.

Fuel Flow

While it is feasible to consider the air flow as unaffected by the presence of the fuel, the converse is decidedly not a valid assumption. The movement of the fuel is almost wholly dependent upon aerodynamic forces imposed by the air, with gravitational forces being a second important influence.

It is generally assumed that the fuel flows in the induction system either as vapor, as entrained liquid droplets, or as a liquid film on the wall.

From Figure 9, it is apparent that fuel is carried from the main venturi by a fraction of the air flow. High-speed movies taken at the University of Michigan in connection with a Masters Thesis by Orlee are said to have shown liquid-fuel streams becoming attached to the venturi wall and tearing off in "globs", particularly at low air velocities, in contrast to the finely divided spray that "atomizers" are assumed to produce.

Impaction of Droplets. Fuel droplets, whatever their form, are first impacted directly on the throttle plate. There is reason to assume that virtually 100 percent of the fuel droplets will be impacted except near wide-open throttle and that there will have been insufficient time at this point for a significant amount of vaporization of the fuel to have taken place. The fuel is re-entrained by the air in leaving the throttle plate, most likely in very coarse droplets. From Figure 9, it can be seen that, under part-throttle conditions, the air-fuel mixture is highly stratified as it leaves the throttle bore. As the mixture turns a 90-degree bend on entering the manifold, virtually all the liquid droplets are again impacted on the manifold floor.

Vaporization. Since it is conventional practice to heat the central part of the manifold floor by exhaust gases, significant vaporization of fuel at the wall occurs. Heat exchange between the air and the liquid film also

helps promote vaporization. It appears that the extent to which vaporization occurs has not been subjected to much study, except by visual observations. Some investigators consider the manifold wall to be dry under road-load cruise conditions--others consider it slightly wet. Under wide-open-throttle conditions, however, there appears to be agreement that a large portion of the fuel does not vaporize but travels in a liquid film along the manifold walls.

Liquid Film. Movies taken by the Ethyl Corporation with an unheated glass manifold on a 6-cylinder engine with dyed fuel show strikingly that the liquid film is wavy and irregular. Other investigators, using windows in conventional manifolds, report the fuel flowing in wavy films or irregular rivulets covering only part of the manifold surfaces under WOT conditions.

Under WOT conditions, the stoichiometric-mixture liquid-fuel-flow rate by volume is only about one-hundredth of one percent of the air flow; therefore, visual observation of substantial liquid films on the wall is an indication of a very large liquid-fuel holdup in the manifold. No accurate measure of the amount of holdup, the film thickness, or the velocity slip ratio appears to be available.

It is probable that some of the liquid is reentrained from the film; however, droplets so entrained probably impact on the next bend.

Stratification and Maldistribution. It can be assumed that the portion of the fuel which is vaporized will follow the air; however, there is a good chance that the vaporized fuel and air do not become thoroughly mixed before they arrive at the intake valves.

In a branched manifold, any stratification of the fuel as it follows the air, can effect separation of the air and fuel streams in traversing a branch, thus causing geometric maldistribution in the manifold. Geometric maldistribution can be expected to be most severe when a large fraction of the fuel is in a liquid film on the manifold wall.

ERRATA SHEET

for Phase Report

A STUDY OF THE INFLUENCE OF FUEL ATOMIZATION,
VAPORIZATION, AND MIXING PROCESSES ON POLLUTANT
EMISSIONS FROM MOTOR-VEHICLE POWERPLANTS

Contract No. CPA 22-69-9

April 30, 1969

- Page 67 - Bottom table, third and fourth column headings should be " ΔT " instead of "T".
- Page 82 - Bottom number on vertical axis should be "1" instead of "0".
- Page 85 - Equation (30) near bottom of page, number "4" should appear as a constant in front of " π^2 ".
- Page 87 - Figure 21, numbers at top and bottom of vertical axis should be " 6×10^4 " instead of " 0.6×10^4 " and " 4×10^2 " instead of " 0.4×10^2 ".
- Page 111 - Figure 28, "40,000 rpm" should be changed to "45,000 rpm".

In a single-cylinder engine, there would be no geometric maldistribution, since all the air and fuel arrives at the same place. However, with stratification of the air and fuel arriving at the intake valve, in-cylinder nonhomogeneity can and apparently does occur. In addition, time variations in the air-fuel ratio will occur, primarily as a result of the irregular flow of the liquid-fuel film. There is also some reason to suspect that the fuel metering at the carburetor is not constant. Since the fuel arrives at the venturi in slugs separated by slugs of air and since the air flow at the venturi is pulsating, there is the possibility of a "beat" phenomena causing cyclic variations in metering. In addition, the fuel attachment to the venturi wall that has been observed could contribute to irregularities in the flow of fuel entering the manifold.

In conventional multicylinder-engine induction systems, the time-variation maldistribution is superimposed on the geometric maldistribution.

Air-Flow Mathematical Models

Plenum Model

A simple way of looking at the intake manifold and associated cylinder-head intake ports is to assume that it is a plenum with uniform pressure throughout, from which air is withdrawn in some periodic manner and into which air flows as a function of the pressure differential at the inlet and the restriction imposed by the throttle. Physically, such a model is illustrated by Figure 11. For such a model, assuming perfect-gas behavior of the air-fuel mixture, the dynamic pressure may be computed by

$$\frac{dp}{dt} = \frac{\gamma p}{V_m} \left[\frac{w_{in}}{\rho_{in}} - \frac{w_{out}}{\rho_{out}} \right] + \frac{(\gamma-1)q}{V_m}^* \quad (1)$$

* Definitions for symbols used in equations are found in the Nomenclature (page 143).

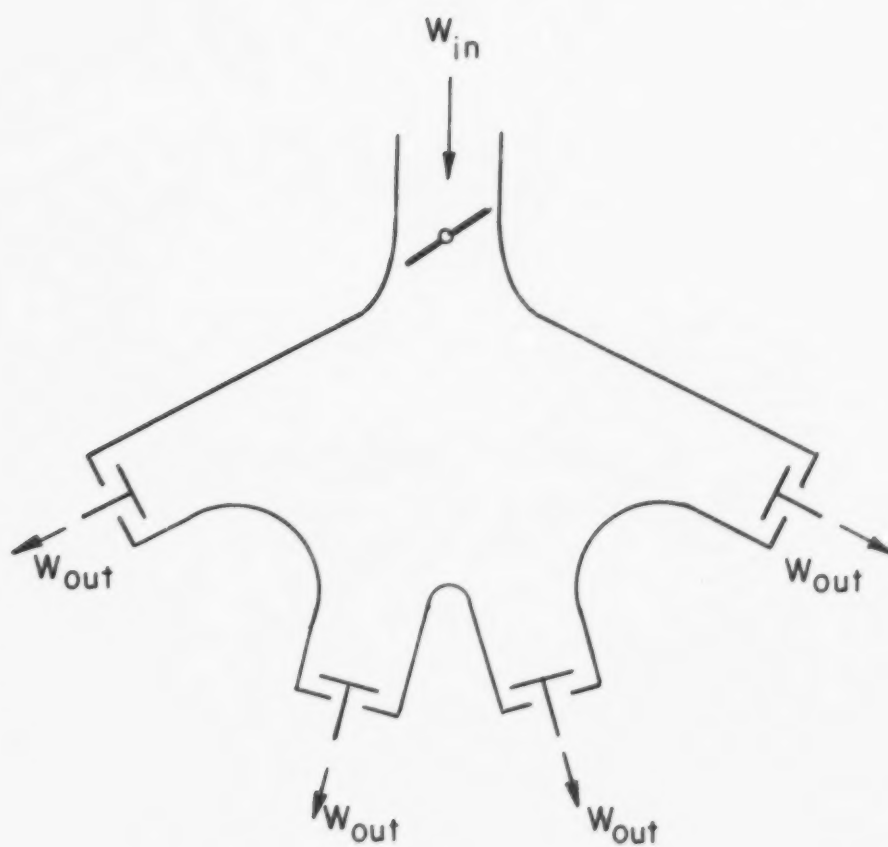


FIGURE 11. PLENUM MODEL OF MULTICYLINDER INTAKE MANIFOLD

A derivation of this expression is presented in Reference (14). For practical purposes, the function for ω_{cvt} can be assumed to be a rectified sine wave, as indicated by Figure 10. Alternatively, approaches to computing the intake-valve flow as a function of the cylinder pressure, such as presented by Huber and Brown⁽¹⁵⁾ or Wolgemuth and Olson⁽¹⁶⁾, can be used if desired. However, there appears to be little incentive for using these more detailed approaches for investigation of intake-manifold phenomena alone.

Textbook expressions for compressible-fluid flow can be used to compute ω_{in} . The heat-transfer rate can be estimated by methods presented in the section of this report on vaporization. The resulting set of equations can be solved easily using electronic digital-computer techniques for solving differential equations.

The plenum model is useful for investigating manifold-pressure fluctuations resulting from volumetric effects and the resulting air-flow fluctuations through the carburetor. It gives no information concerning the local pressure and flow field in the manifold and gives no consideration to momentum effects.

Straight-Pipe Model

Dynamic effects in intake manifolds have been investigated by assuming the intake manifold to be a straight pipe of uniform cross-sectional area. Studies conducted have been for single-cylinder models; thus, the effect of the branches in multicylinder-engine manifolds has not been accounted for. Writing continuity, momentum, and energy-balance equations for one-dimensional unsteady flow produces the following expressions presented by Goyal, Scharpf, and Borman⁽¹⁷⁾:

$$\frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial x} + u \frac{\partial \rho}{\partial x} = 0 \quad , \quad (2a)$$

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{1}{\rho} \frac{\partial p}{\partial x} + F_f = 0 \quad , \quad (2b)$$

$$(\gamma-1)p(q + uF_f) = \frac{\partial p}{\partial t} + u \frac{\partial p}{\partial x} - c^2 \left(\frac{\partial \rho}{\partial t} + u \frac{\partial \rho}{\partial x} \right) \quad , \quad (2c)$$

where

$$F_f = \frac{4f}{D} \frac{u^2}{2} \frac{u}{|u|} \quad (2d)$$

Goyal, et al, used these equations to investigate an unthrottled, single-cylinder-engine intake and exhaust system. In this study, friction and heat transfer were neglected in the intake system, in which case, the equations simplify to

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{c^2}{\rho} \frac{\partial \rho}{\partial x} = 0 \quad (3a)$$

$$\frac{\partial \rho}{\partial t} + u \frac{\partial \rho}{\partial x} + \rho \frac{\partial u}{\partial x} = 0 \quad (3b)$$

McAulay, et al⁽¹⁸⁾, also used these equations as part of a simulation of a single-cylinder compression-ignition engine.

These equations can be expressed in a single equation in terms of a velocity potential defined as

$$u = \partial \varphi / \partial x \quad (4)$$

The resulting expression, as presented by Shapiro⁽¹⁹⁾, is

$$\left[c^2 - \left(\frac{\partial \varphi}{\partial x} \right)^2 \right] \frac{\partial^2 \varphi}{\partial x^2} - 2 \frac{\partial \varphi}{\partial x} \frac{\partial^2 \varphi}{\partial x \partial t} - \frac{\partial^2 \varphi}{\partial t^2} = 0 \quad (5)$$

This form was used by Tsai⁽²⁰⁾, and Taylor, Livengood, and Tsai⁽²¹⁾ in studies of inlet systems of unthrottled, single-cylinder engines.

In all of these cases, the method of characteristics was used to obtain a solution to the differential equations. Necessary to the solution are the assumptions of boundary conditions: the rate of flow as a function of time at the intake valve and an expression for the pressure at the pipe inlet. Practical use of this mathematical model can be made only with the aid of electronic digital computers, and the methods of solution are not simple.

The result of these studies, in most cases, has been the computation of the dynamic pressure at the intake valve, which is of interest because of its effect on volumetric efficiency. Besides modeling the "ram"-charging effects of the inlet pipe, the results also effectively show the "organ-pipe" oscillations that occur after valve closing. At present, this approach appears to be too complex to be used as a design tool in engineering intake systems for throttled multicylinder engines.

A much simpler but less accurate model can be used to investigate inertial effects in the manifold. The column of air in the manifold, extending from the throttle bore to the intake valve, can be viewed as a slug of air having uniform density, velocity, and cross-sectional area; the pressure gradient in the column due to inertial effects can be estimated as

$$\frac{dp}{dx} = \frac{\rho}{g_c} \frac{du}{dt} \quad (6)$$

This expression can be viewed as a direct simplification of the Navier-Stokes equations, as presented by Schlichting⁽²²⁾, in which one-dimensional flow is assumed and body and viscous forces are ignored. This expression is practical for use with hand-calculation methods; it would provide an approximation of the "ram" effects in the manifold but does not yield information concerning local velocities and pressures, nor will it predict any resonant acoustic phenomena when used by itself.

Helmholtz-Resonator Model

Thompson and Engelman⁽²³⁾ have proposed viewing the induction system and engine cylinder as a Helmholtz resonator, as illustrated by Figure 12. A Helmholtz resonator may be thought of as a spring-mass system in which the air in the cylinder acts as a pneumatic spring and the air in the pipe acts as the mass. If the effective cylinder volume is taken with the piston at mid-stroke, the natural frequency of the system will be computed by

$$f = \frac{c}{2\pi} \sqrt{\frac{2A_{x,m}}{V_0 L_m}} \sqrt{\frac{\gamma_c - 1}{\gamma_c + 1}} \quad (7)$$

Thompson and Engelman show that there is a relationship between the Helmholtz frequency and engine speed at which maximum charging will occur. This is a noninteger relationship; true resonant phenomena of the Helmholtz type are prevented from occurring, since the closing of the intake valve "shuts off" the resonator a short time after the oscillation begins.

This approach does not provide detailed information on local pressures and velocities; its main usefulness is to provide simple rules for designing manifolds for favorable "ram"-charging characteristics.

Organ-Pipe Model

As previously pointed out, organ-pipe tuning is not used to achieve favorable charging in conventional passenger-car engines; however, organ-pipe phenomena do occur. It has also been pointed out that a detailed solution of the straight-pipe continuity, momentum, and energy equations will show organ-pipe phenomena. Computed data exhibiting this are presented by McAulay, et al.⁽¹⁸⁾ Experimental pressure traces are presented by Thompson and Engelman.⁽²³⁾

Organ-pipe tuning can be investigated independently using the one-dimensional wave equation,

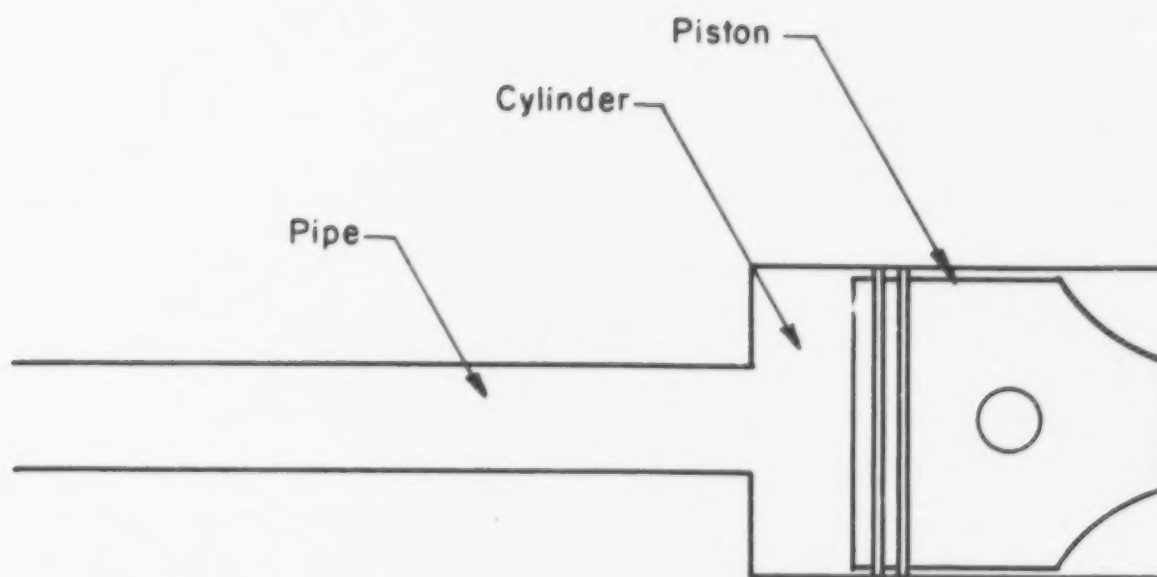


FIGURE 12. HELMHOLTZ-RESONATOR MODEL OF INTAKE MANIFOLD

$$\frac{\partial^2 p}{\partial x^2} = \frac{1}{c^2} \frac{\partial^2 p}{\partial t^2} \quad (8)$$

which is valid only for small amplitudes of pressure variation and which has the general solution

$$p = F_1\left(t - \frac{x}{c}\right) + F_2\left(t + \frac{x}{c}\right) \quad (9)$$

where F_1 and F_2 are arbitrary functions which depend upon the initial conditions. This approach could be used to predict acoustic pressure variations in an inlet pipe, given a set of initial conditions. Computation methods are usually lengthy and complex.

A much simpler approach is to compute the natural frequencies by the relationship

$$f = \frac{mc}{2L}, \quad m = 1, 2, 3, 4, \dots \quad (10)$$

This will not yield information on the amplitude of the oscillations.

Composite Model

It does not appear that a composite mathematical model has been achieved that will successfully predict the principal phenomena that occur in a throttled, multicylinder induction system; in fact, we have no evidence that it has been attempted. Mathematically, it appears possible, but the computational difficulties would be formidable.

The main incentive for assembling such a model for the purposes of this study is to characterize the air flow and pressure--that is, to provide a map of the flow and pressure in the manifold for successive increments of time throughout one complete engine cycle. This information would aid in understanding the nature of the aerodynamic forces on the liquid fuel droplets and wall film and in understanding better how and when geometric maldistribution of the fuel occurs. Much of this understanding can be gained by the individual use of several simpler mathematical models.

Fuel-Droplet Dynamics

Of considerable interest in the design of induction systems is the transport of the fuel from the carburetor to each cylinder. The fuel may exist as a vapor, as droplets, or as a film on the walls of the intake manifold. The vaporized portion of the fuel and the droplets which remain suspended in the air can be expected to be transported satisfactorily. Liquid on the walls of the manifold will tend to result in nonuniform flow of fuel to individual cylinders because of the surging and wave-like nature of liquid flow over wall surfaces as induced by air flow over the surface of the liquid. Hence, it becomes important to consider the transport of liquid droplets through an induction system and, more specifically, the tendency of suspended drops to deposit from the air-fuel mixture flow and form a liquid film on the induction-system walls.

Droplet Deposition Mechanisms

The deposition of liquid drops from a flowing gas stream may occur by a number of different mechanisms. These mechanisms reflect the forces which can be exerted on suspended droplets, causing them to have a motion relative to the suspending gas. The mechanisms of deposition can be enumerated as inertial impaction, Brownian diffusion, turbulent diffusion, thermophoresis, diffusiophoresis, and electrophoresis.

Of these mechanisms, the only one of particular significance for an induction system is inertial impaction. Inertial impaction may result from

the acceleration of the gas-droplet mixture either in a rectilinear or curvilinear manner. Inertial impaction, for example, would include gravitational settling of droplets and deposition of droplets during angular acceleration of the gas through a bend in the manifold passages. Because of the short length of time between atomization of the liquid and arrival at the intake port of the cylinder, gravitational settling is expected to be of minor importance. However, the high gas velocities which occur and the necessity for bends in the manifold would indicate that inertial deposition of liquid droplets from curvilinear flow may be of significant importance.

Inertial Impaction of Droplets

Inertial impaction of droplets in bends and on obstacles can be described through an understanding of the trajectories of droplets in curvilinear flows. Basically, the deposition process proceeds because, as the streamlines of the flow turn at a bend, the greater inertia of the droplets (as compared to the carrier gas) causes them to move across the streamlines and impact on the collecting surface. In the case of flow through a pipe bend, the outside pipe wall at the bend becomes the collecting surface.

The analysis of inertial deposition of droplets is based on an equivalence of drag and acceleration forces. This can be expressed as follows: ⁽²⁴⁾

$$(\vec{u}_d - \vec{u}_g) |\vec{u}_d - \vec{u}_g| \left(\frac{\pi \rho_g D_d^2 C_D}{8 g_c} \right) = - \left(\frac{\pi \rho_d D_d^3}{6 g_c} \right) \frac{d\vec{u}}{dt} \quad (11)$$

The usual procedure required for analytically predicting the motion of a droplet is based on prior knowledge of the velocity field for the carrier gas and involves a step-wise calculation of the drop trajectory. If the drop trajectory intercepts the collecting surface, impaction is said to occur.

It is evident that the fate of any droplet is dependent on its location in the gas stream upstream from the bend. This prior location determines whether or not a particular droplet will impact; the fraction of the droplets of a single size impacted is referred to as the impaction efficiency. This efficiency can be determined by multiple calculations of droplet trajectories.

A dimensional analysis indicates that three dimensionless groups control droplet impaction. These are

- Impaction parameter

$$K = (2\pi n)^{1/2} \left(\frac{\rho_d u_g}{18 \mu_g D_m} \right)^{1/2} D_d \quad (12)$$

- Parameter characterizing non-Stokes law behavior

$$\Phi = \frac{9 \rho_g^2 u_g D_m}{\rho_d \mu_g} \quad (13)$$

- Geometry of bend

$$R_b / D_m \quad (14)$$

The Reynolds number for the gas flow is not included in the above groups when the gas flow, as described by Ranz⁽²⁴⁾, is ideal and the velocity vector is not a function of Reynolds number of the gas flow. This case has been analyzed by Ranz, and the results have been presented as a dimensionless correlation of impaction efficiency in circular ducts as a function of the impaction parameter, K , for various values of the combined parameter $\Phi / (2\pi n)^2 (R_b / D_m)^2$.

Using this dimensionless correlation, droplet impaction was calculated for this study as dependent on droplet size, gas-flow rate, and duct geometry

for several configurations representing the manifold section between the hot spot and the cylinder intake port. It is important to note that the analysis of Ranz considers only the bend itself and does not consider deposition in any duct section beyond the bend, although such deposition may result from droplet velocities attained in the bend.

Figure 13 shows the geometries considered for impaction calculations. Geometry A is a sharp 90-degree bend with a 1-1/2-inch bend radius. Geometry B is also a 90-degree bend but with a larger 6-inch radius. Geometry C is a 30-degree bend with a 6-inch radius, and Geometry D is a 30-degree bend with a 12-inch radius. Geometry E is similar to B with the 6-inch radius but with a 33 percent larger passage diameter. Geometries A and C have straight sections on either end of the curved section. Deposition in these straight sections is not included in the Ranz analysis.

Figure 14 shows the results of impaction calculations for Geometry A. Gas-flow rates of 48, 79, 119, and 143 cfm were used which correspond to flow velocities of 65, 108, 162, and 194 fps, respectively. Only droplet sizes below 40 microns were assumed, because in a practical situation larger drops would most likely be impacted before entering the manifold section under consideration. These curves show that the fraction impacted increases with gas-flow rate and with droplet size. A similar pattern would apply to all geometries.

Figure 15 shows the impaction for all five manifold geometries at a single gas-flow rate of 79 cfm. This flow rate would correspond approximately to medium-speed operation. The curves are identified according to the manifold geometry represented.

For the small radius, 90-degree bend (Geometry A), the analysis predicts that fewer droplets are impacted than for the larger radius, 90-degree bend (Geometry B). The reason that the sharp bend removes less droplets than the sweeping bend is that the more rapid acceleration of the droplets in the sharp bend causes the flow regime for the droplets to be non-Stokesian such that the drag coefficient is greater. The increased drag results in less deflection of the droplets from the air stream line for the same amount of

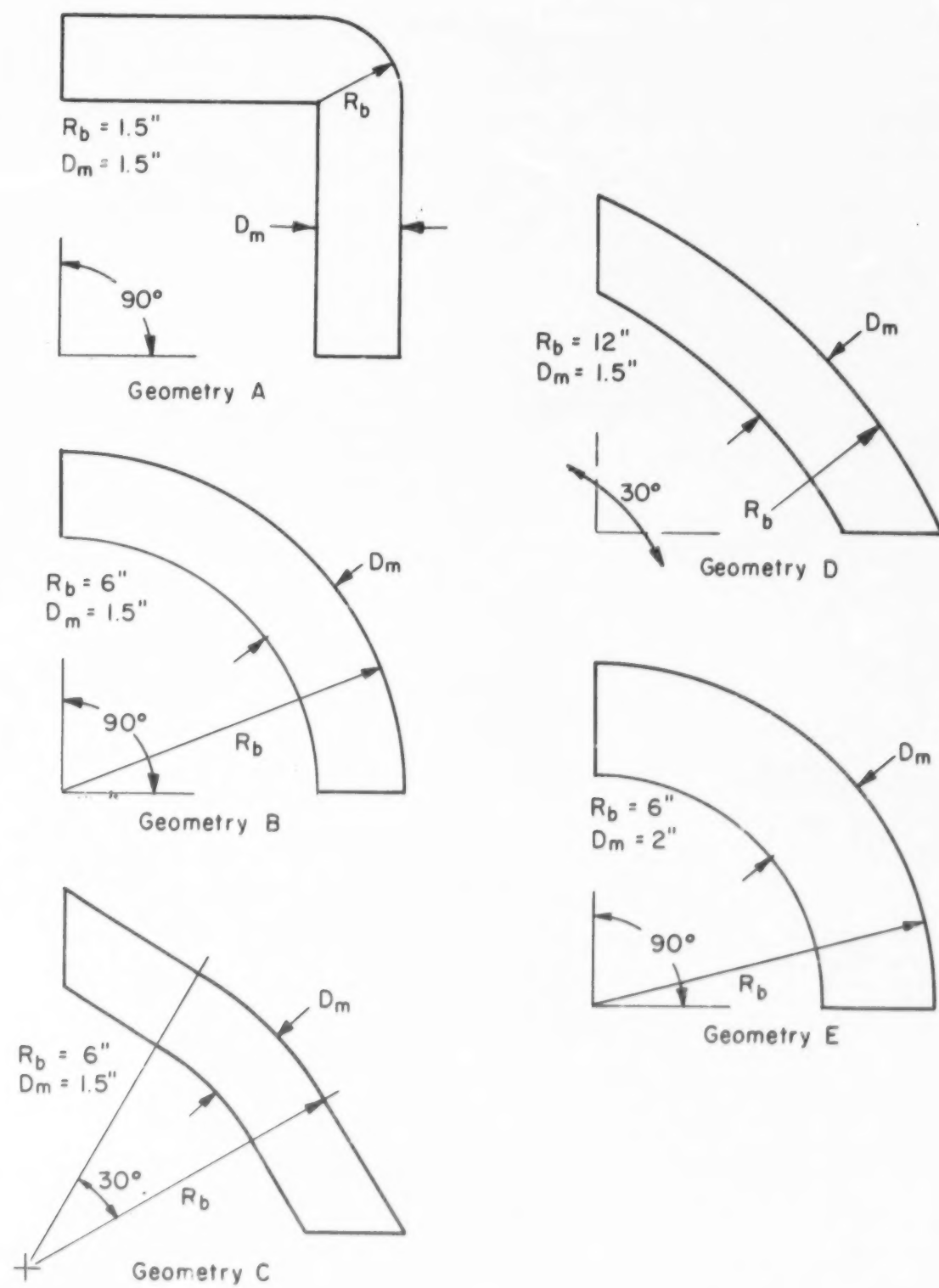


FIGURE 13. CIRCULAR-BEND GEOMETRIES FOR IMPACTION CALCULATIONS

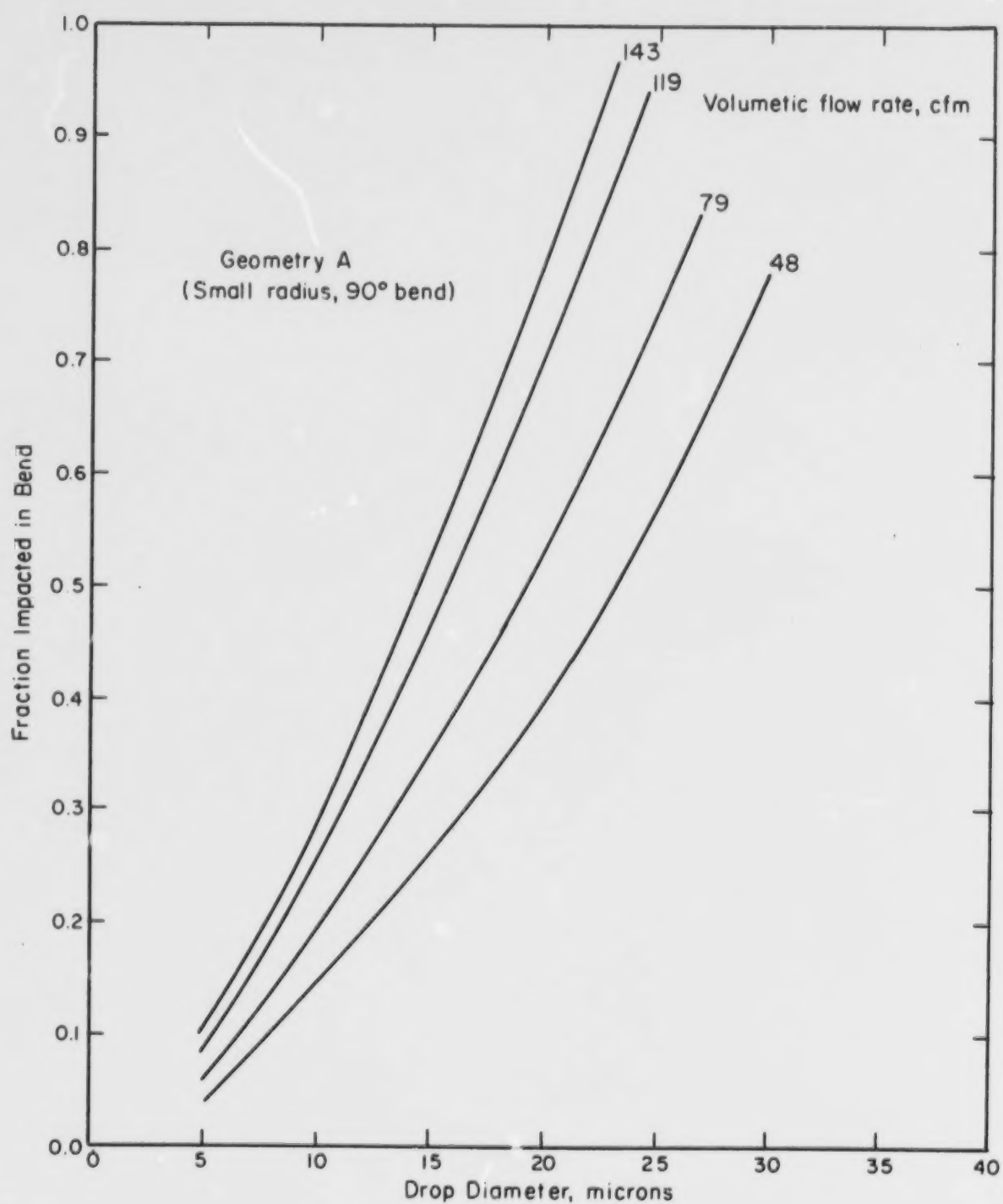


FIGURE 14. ESTIMATED FUEL-DROPLET IMPACTION IN 90-DEGREE CIRCULAR BEND WITH SMALL RADIUS

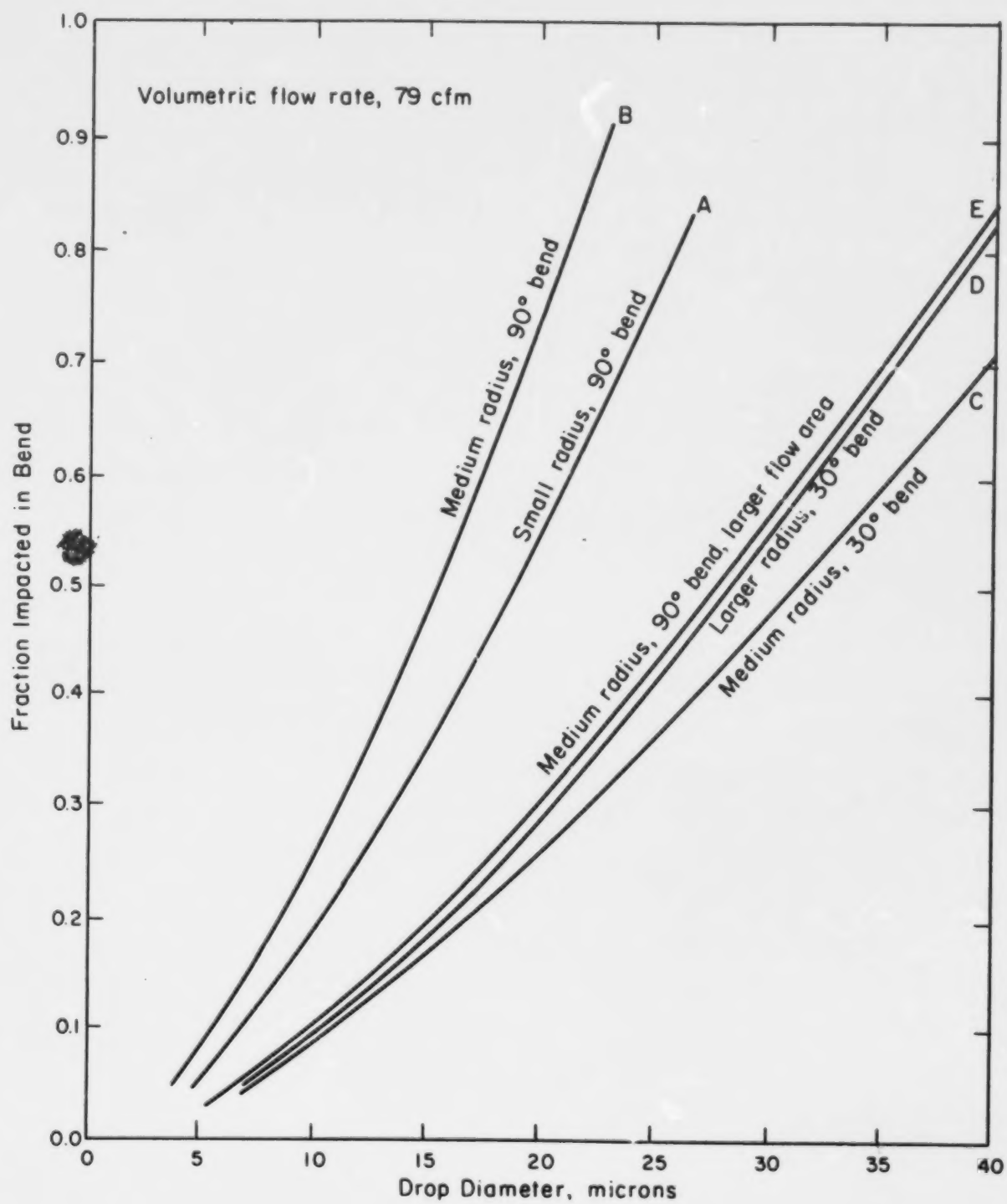


FIGURE 15. ESTIMATED FUEL-DROPLET IMPACTION IN VARIOUS CIRCULAR BENDS

angular acceleration. However, if estimates are made for the deposition in a straight section beyond each bend and caused by the droplet motion induced by the bend, a greater fraction of additional droplets would be impacted after the bend having the small radius. These estimates of deposition beyond the bend are based on calculation of the distance required to stop the droplet, relative to the gas, by drag forces when the droplets are introduced at some terminal velocity. The droplet terminal velocity is set by the drag forces and acceleration in the bend.

Deposition beyond the bend increases with gas velocity and droplet size, as does deposition within the bend. However, deposition beyond the bend decreases with increasing radius of curvature in contrast to these effects within the bend. Hence, the effect of bend radius is not shown completely in Figure 15, which considers the bend only.

When the gas stream is turned through a smaller angle (Geometries C and D), there is less droplet impaction, and when the passage diameter is increased for a given mass flow (Geometry E), impaction is also reduced. There are two reasons for this latter effect, which are illustrated by comparing Geometries B and E. First, the passage being larger, the droplets have a greater distance to move before hitting the wall of the bend and, second, the gas velocity is lower and reduces the inertia of the droplets.

Considerations of deposition beyond a bend introduce complexities which indicate that, for a more adequate analysis of deposition in an intake manifold, the droplet trajectories should be calculated for the entire system and not just the bends. For example, even though a droplet has a radial velocity induced by a bend, the time available for the droplet to reach the wall is also a factor and depends on the position of the bend in the manifold. In spite of the fact that complete trajectory calculations could be made, they are probably not justified because of the idealized flows which must be assumed, including steady flow and the absence of flow separation at the bend throat. It seems more direct, and certainly more definitive, to perform experimental measurements of deposition systems which are designed using known principles for reducing deposition through geometry considerations.

Several generalizations concerning geometry effects can be drawn from the analyses and impaction calculations. The following comments suggest geometry considerations directed toward reducing impaction in bends.

- (1) Bends through the minimum angle required should be used.
- (2) Passage diameters should be as large as practicable.
- (3) If cross-sectional area must be constant or is fixed to maintain a specified gas velocity, the possibility of flattening the cross-sectional shape in the bend should be considered such that the elongated axis is in the plane of the bend angle.

Although the impaction characteristics of circular bends presented in Figures 14 and 15 are probably optimistic, they do serve to indicate that a major fraction of droplets in the size range of 10 to 20 microns could pass through an intake manifold without impaction.

Droplet Impaction in a Conventional Induction System

An example calculation was made to investigate the probable droplet-impaction characteristics of present conventional induction systems. For this example, a triple-venturi conventional carburetor was assumed to discharge into a 90-degree manifold bend, and the fraction of the fuel droplets impacted in the bend was computed. Fuel vaporization and the usual effects of the presence of the throttle plate were ignored.

The gas velocity through the venturi was computed from pressure measurements supplied by Holley Carburetor Company for this type of carburetor. The velocity in the bend is computed from the volumetric flow rate, with the assumption that the bend cross-section diameter is 1.68 inches. Droplet size is computed using empirical relationships for two-fluid atomizers presented in a later section of this report, and the fraction impacted is computed with methods described in the immediately preceding section.

The results of the example calculations are presented in the following tabulation and in Figure 16.

Volumetric Air Flow Rate, cfm	Venturi Air Velocity, ft/sec	Air Velocity in Bend, ft/sec	Estimated Droplet Diameter, microns	Fraction of Fuel Droplets Impacted
60	134	65	76	1.00
100	307	108	33	1.00
150	482	162	21	0.68
180	603	194	17	0.56

From these results, it is shown that the droplet size produced by venturi carburetors decreases substantially as air flow is increased. The higher air velocities are conducive to greater impaction, but the effect of reduced droplet size predominates. As a result, we can estimate that, in current induction systems, virtually all of the fuel droplets produced by the carburetor are impacted in the first manifold bend at low air flows. At higher air flows, a substantial fraction of the droplets would ideally stay entrained. However, non-ideal performance of the venturi atomizer and the effect of the throttle plate probably induce greater impaction in real systems than indicated by the values computed in this example.

Also plotted in Figure 16 is the fraction of droplets impacted if the carburetor could produce 17-micron droplets over the complete flow range. This is a striking illustration of the improvement in droplet transport that could be obtained with an improved atomizing device.

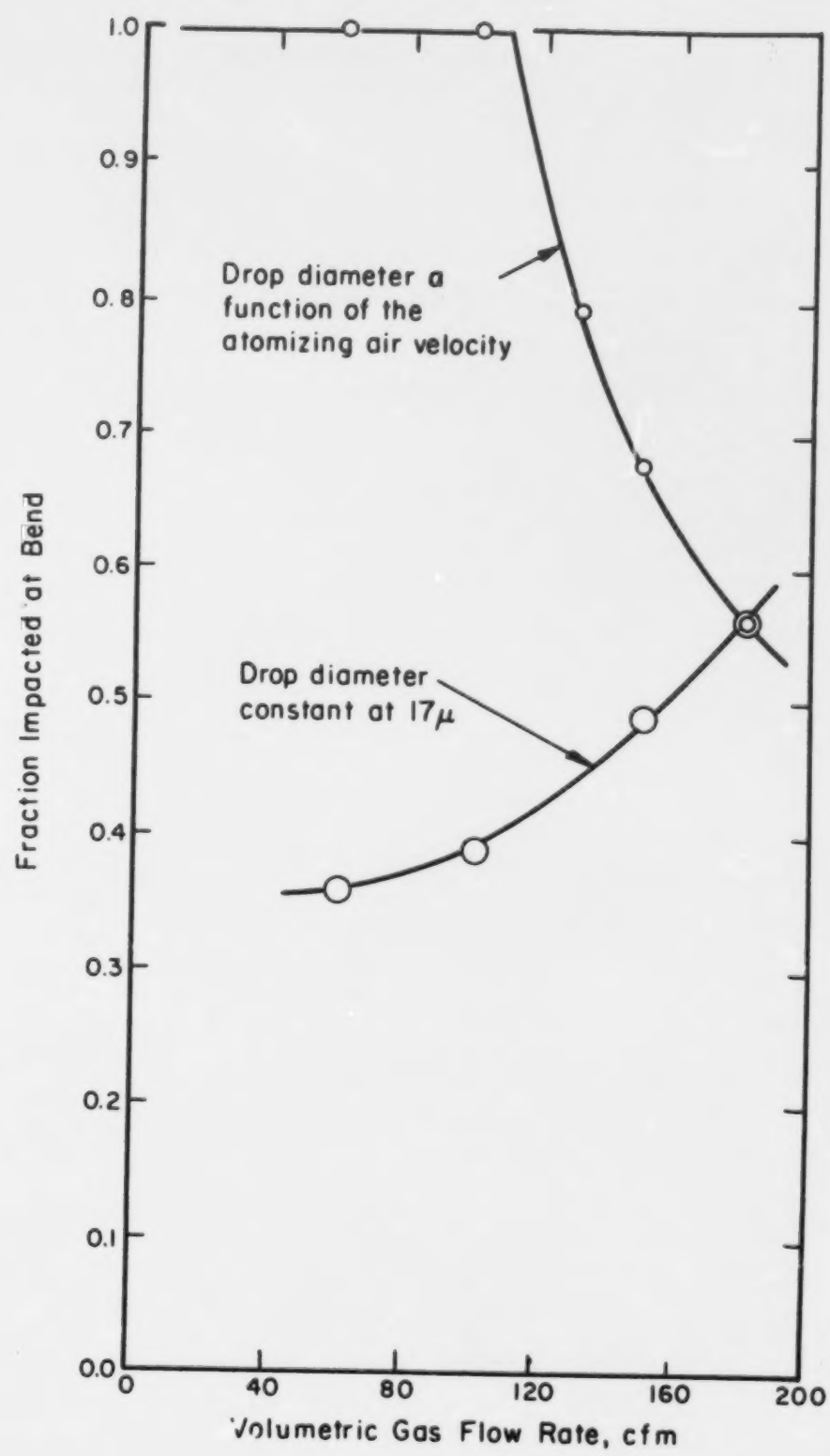


FIGURE 16. ESTIMATED FUEL-DROPLET IMPACTION IN THE FIRST MANIFOLD BEND WITH CONVENTIONAL CARBURETION

Vaporization and Heat Transfer

Vaporization of fuel in an induction system may occur either from droplets of fuel suspended in the air or from the film of fuel on the walls. Vaporization requires that the latent heat of the fuel, which is about 150 Btu/lb for an average gasoline fraction, be supplied from some source: sensible heat of the fuel or air (which may be augmented by an air-preheating device) or heat transferred to the fuel from the intake-manifold walls.

Equilibrium Vaporization

The minimum temperature at which the fuel can exist entirely as vapor in equilibrium with the combustion air, assuming perfect mixing, is the dew point of the mixture. The dew point depends upon the concentrations and vapor pressures of the hydrocarbons making up the fuel, the air-fuel ratio, and the total pressure.

An ASTM distillation curve is commonly used to characterize gasoline volatility. Specific distillation curves will vary with the geographical area in which the gasoline is sold, the season of the year, and the brand.

Figure 17 shows a gasoline distillation curve which is an average of data for brands, districts, and summer and winter grades obtained from Bureau of Mines publications.^(25,26) The ASTM curve gives the boiling point of the remaining mixture in the presence of its own vapor as a function of the percent evaporated.

Empirical methods are available for estimating the dew point of the air-fuel mixture from the ASTM distillation curve. Using procedures of Edmister and Pollock⁽²⁷⁾ and the data in Figure 17, the following approximate dew points can be computed:

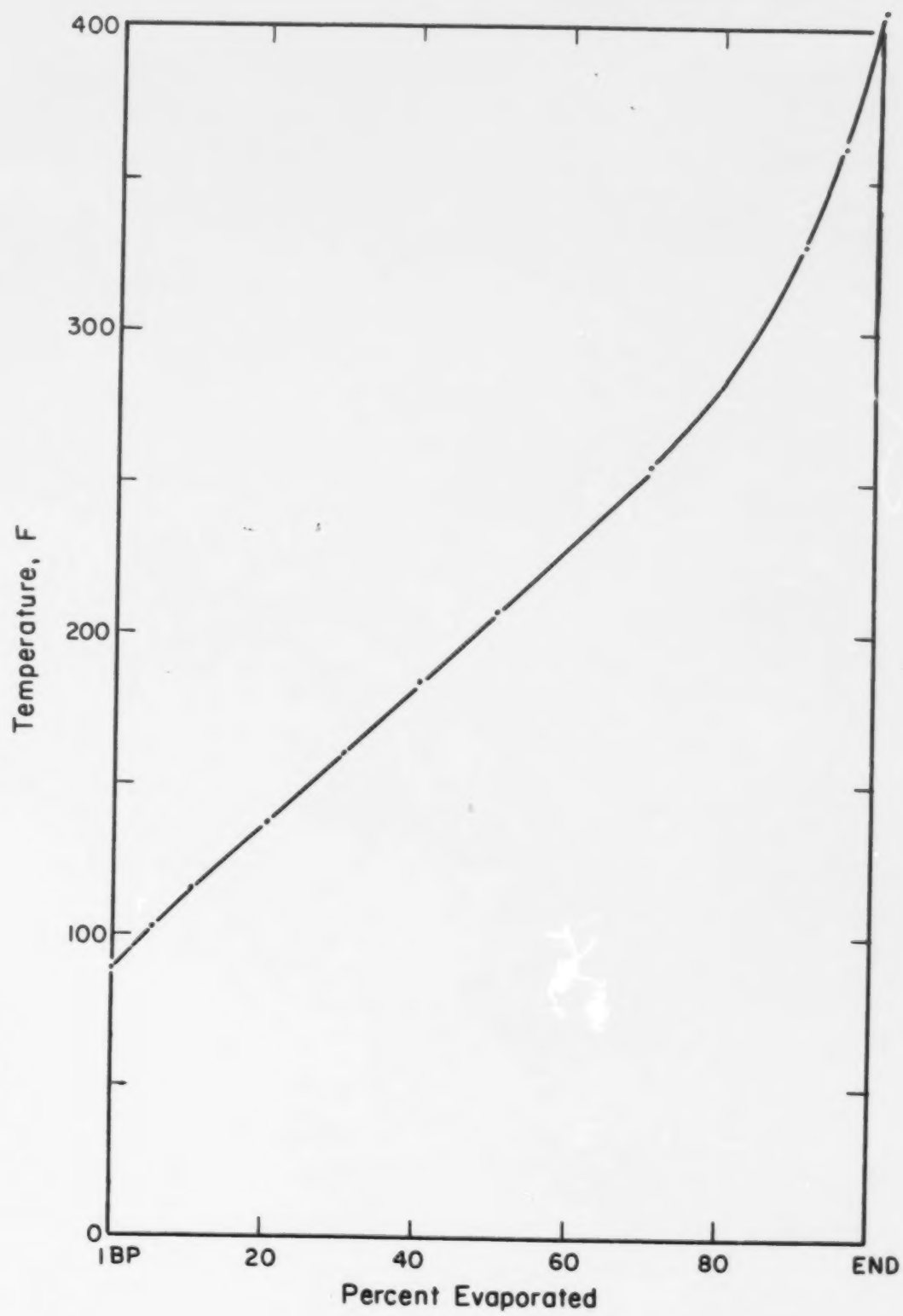


FIGURE 17. DISTILLATION CURVE FOR A COMPOSITE GASOLINE

BATTELLE MEMORIAL INSTITUTE - COLUMBUS LABORATORIES

Mixture Pressure, in. Hg Vacuum	Dew Point of Gasoline-Air Mixture, F	
	15:1 Air-Fuel Ratio	20:1 Air-Fuel Ratio
0	116	105
2	113	103
10	101	92
15	93	84
20	80	71

The above tabulation shows that the dew point is lowered by both reduced total mixture pressure and leaner mixture concentration--both have the effect of reducing the partial pressure of the hydrocarbon vapor. These data also show that fairly warm mixture temperatures are required if the gasoline is to be completely vaporized. If intake-manifold mixture temperatures in the range of 120 to 160 F are taken as typical for warmed-up operation, it is possible for all the fuel to be vaporized if the mixture is given sufficient time to reach thermal equilibrium. Actually, the time is not sufficient, as will be discussed subsequently.

At lower mixture temperatures, it will not be possible to vaporize the fuel completely. Using Bridgeman's charts⁽²⁸⁾ and the ASTM distillation curve from Figure 17, the percent of the fuel vaporized at equilibrium conditions as a function of mixture temperature can be computed.

Figure 18 shows these relations for a 16:1 air-fuel ratio at one atmosphere and at 0.5 atmosphere mixture pressures. The large amount vaporized at below-zero (F) temperatures is due principally to the lighter hydrocarbons, which typically range as low as C₄. Figure 18 shows that about half of the fuel can be vaporized at equilibrium in a 16:1 air-fuel ratio mixture during cold-winter-starting conditions.

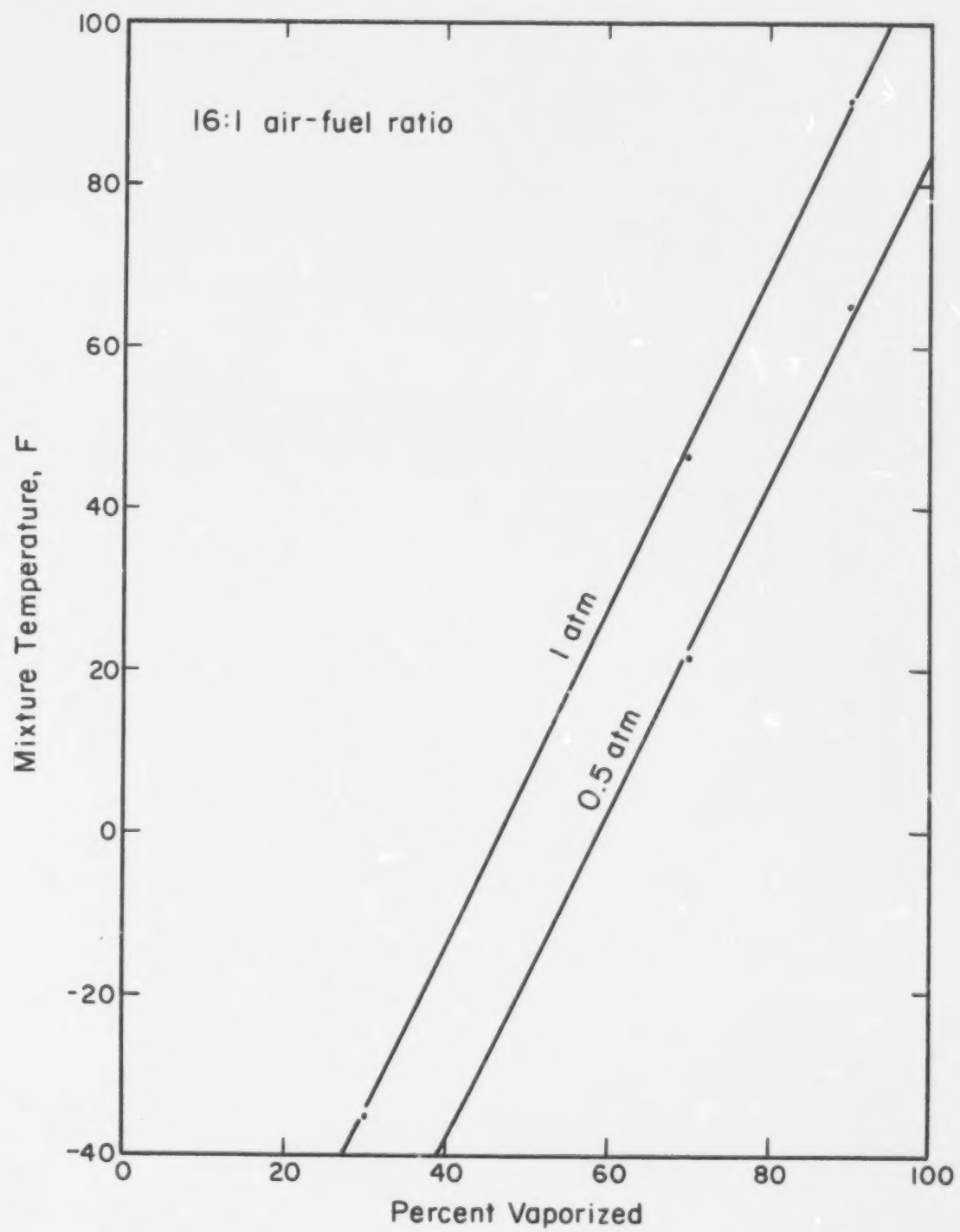


FIGURE 18. EQUILIBRIUM VAPORIZATION OF GASOLINE FUEL IN AIR

Droplet Vaporization

The problem of calculating the complete history of an evaporating droplet carried in an air stream is a difficult one, even though the separate phenomena are understood reasonably well. The droplet drag, mass transfer and heat transfer all mutually interact, and complete solutions apparently require the use of electronic-computer calculation. El Wakil, et al⁽²⁹⁾, have carried out such calculations for droplets of n-octane, benzene, and pentane for a few selected initial conditions.

To set the problem in focus, the time required to evaporate a droplet, considering only the heat transfer problem, will be calculated. The rate of change of mass of the drop is

$$\frac{dm}{dt} = - \frac{hA_d \Delta T}{\lambda} \quad (15)$$

where λ is the latent heat. Assuming the drop is spherical, expressions for mass and area substituted in the above expression give the following

$$\rho_d D_d^2 \frac{dD_d}{dt} = - \frac{\pi k_d D_d \Delta T}{\lambda} \quad (16)$$

The Reynolds number is assumed sufficiently low that $N_{Nu} = hD/k = 2$, a constant. Integrating between the limits of the initial diameter, D_d , and zero gives

$$D_d^2 = \left(\frac{8 k_d \Delta T}{\lambda \rho_d} \right) \Delta t \quad (17)$$

Δt being the time required to evaporate the droplet. The following tabulation lists the necessary values of temperature difference, ΔT , for two values of the time interval, Δt , 10 and 100 milliseconds, assuming the fuel has a latent heat of 150 Btu/lb and a specific gravity of 0.684, essentially equivalent to n-heptane.

Droplet Diameter, microns	Temperature Difference Required to Evaporate a Droplet, F	
	t = 10 millisec	t = 100 millisec
10.0	17.1	1.71
31.6	171.0	17.1
100.0	1,710.0	171.0
316.0	17,100.0	1,710.0

These combinations of times and temperature appear to agree reasonably well with the results for n-octane calculated by El Wakil, et al⁽²⁹⁾.

If it is assumed, as a rough approximation, that one cylinder intake stroke loads the manifold with a cylinder full of mixture and the next intake stroke pulls it into the cylinder, it can be assumed that the residence time of most droplets is the duration of one engine cycle, or two revolutions. However, in a V-8 manifold, this would be correct for a branch, but the runner receives two flow pulses per engine cycle, or one per revolution. Also, some of the droplets will traverse the manifold during one intake stroke, or half a revolution. The uneven sequencing of intake pulses in half of the V-8 intake manifold, and the effects of slip between the droplets and the air cause further variations in residence time. On the average, it appears that the representative residence time is about one engine revolution. With this assumption, 10 milliseconds corresponds to an engine speed of 6000 rpm and 100 milliseconds to a speed of 600 rpm.

It is apparent from the tabulation above that only droplets on the order of 10 microns will possibly have time to vaporize at maximum speed. At idle, 30-micron droplets could vaporize. A 100-micron droplet could not vaporize completely at any speed.

In present induction systems, it appears that none of the droplets stay entrained at idle, and only a fraction can potentially stay entrained at maximum engine speed. Therefore, it can be concluded that droplet evaporation is not a significant process in present systems.

Manifold Wall-to-Air Heat Exchange

The heat necessary to accomplish the vaporization of droplets may be assumed to be transferred to the air from the walls of the induction system. To calculate a heat transfer coefficient, a typical V-8 manifold with a passage length of about 12 inches and with two 90-degree bends was approximated as a uniformly curved pipe 12-inches long with an internal diameter of 1.38 inches.

Steady-flow heat-transfer correlations presented by McAdams⁽³⁰⁾ can be applied to this configuration. The correlation for fully developed turbulent flow,

$$N_{Nu} = 0.023 (N_{Re})^{0.8} (N_{Pr})^{0.33} \quad (18)$$

can be corrected for entrance effects and duct curvature to obtain

$$\frac{h}{G C_p} = 0.023 \left(1 + \frac{7 D_m}{L}\right) \left(1 + \frac{3.5 D_m}{2 R_b}\right) (N_{Re})^{-0.2} (N_{Pr})^{-0.67} \quad (19)$$

Using an arbitrary schedule of intake-manifold pressure and fuel flow to approximate a range of operating conditions, assuming the air flow to be steady, and assuming a constant 15:1 air-fuel ratio, the computation of the temperature difference between the wall and the mixture required to add sufficient heat to the air to vaporize the fuel droplets produces the following results:

Manifold Vacuum, in. Hg	Fuel Flow, lbm/hr	Heat Transfer Coefficient, Btu/hr-F-ft ²	Heat Required Per Branch, Btu/hr	Required Temp. Difference, F
20	4.0	5.86	144	68
15	23.7	25.2	865	95
10	43.5	39.8	1590	110
2	75.0	62.0	2740	122

It is assumed, in the above calculations, that the flow of air is isothermal and that the heat transferred from the wall to the air is in turn transferred to the droplet at the same rate. If all of the latent heat necessary to vaporize the fuel were supplied adiabatically as sensible heat from the combustion air, the initial air temperature would have to be about 40 F higher than the final mixture temperature.

If it is further assumed that 10-micron fuel droplets are evaporating at the dew point of the mixture and that about a 10 F temperature difference between the air and the droplet is required for vaporization, the required wall temperature for vaporization of the droplets can be computed as follows:

Manifold Vacuum, in. Hg	Mixture Dew Point, F	T, Air to 10-Micron Droplet, F	T, Air to Wall, F	Required Wall Temp., F
20	80	10	68	158
15	93	10	95	198
10	101	10	110	231
2	113	10	122	245

The above results indicate that sufficient heat could be added to the air to vaporize all the fuel in 10-micron droplets, considering that current hot-spot

temperatures are in the order of 300 F to 400 F and that the rest of the manifold is probably around 200 F.

Droplets of 30-micron size would require manifold temperatures on the order of 100 F higher; hence, it would probably not be possible to transfer sufficient heat to the air to vaporize completely the fuel droplets in this size range under all operating conditions.

Vaporization from Surface Films

If the fuel is deposited on the induction-system walls, the heat needed for vaporization can be transferred directly from the wall to the fuel without the intervening step of heating the air. Calculation of the vaporization rate from a wetted wall requires in principle that the details of the flow of fuel on the wall be known. As such information is not available, a simple approximation can be made. The fuel can be assumed to flow in a film over a fraction, λ_A , of the manifold wall area at a fraction, λ_d , of the air speed. These two parameters, the area ratio, λ_A , and the slip factor, λ_d , permit the average thickness of the fuel film, the residence time of the fuel in the induction system, and the inventory or holdup of fuel in the induction system to be calculated. Thus, the maximum film thickness, δ is

$$\delta = \frac{w_f}{\pi \lambda_A \lambda_d \rho_f \bar{u}_g D_m} \quad (20)$$

For the schedule of engine operating conditions used in the preceding discussion of heat transfer to the air, a film thickness of about 0.005 to 0.013 in. is computed, assuming all the fuel is on the manifold wall, 0.3 of the wall is covered, and the slip factor is 0.03.

The average residence time, τ , is given by

$$\tau = \frac{L}{\bar{u}_g} \left[\frac{1}{2\lambda_d} + 1 \right] \quad (21)$$

where L is the wetted length. The holdup, V_h , is

$$V_h = w_L \tau / \rho_L \quad (22)$$

The slip factor, λ_L , may thus alternatively be considered as a measure of the residence time and holdup of fuel in the induction system.

The following table lists values of fuel residence time and fuel holdup in one passage of a V-8 manifold as a function of a slip factor for various operating conditions; assuming the last bit of fuel is vaporized just short of the intake valve:

Fuel Residence Time and Fuel Holdup in One Manifold Branch

Slip Factor, λ_L	Vacuum 20 in. Hg Fuel Flow 4.0 lb/hr		15 in. Hg 23.7 lb/hr		10 in. Hg 43.5 lb/hr		2 in. Hg 75 lb/hr	
	τ , sec	V_h , cm ³	τ , sec	V_h , cm ³	τ , sec	V_h , cm ³	τ , sec	V_h , cm ³
1.0	0.055	0.101	0.014	0.015	0.010	0.020	0.008	0.028
0.5	0.082	0.015	0.021	0.022	0.015	0.030	0.012	0.042
0.1	0.30	0.054	0.075	0.082	0.055	0.11	0.044	0.154
0.05	0.57	0.10	0.14	0.16	0.10	0.21	0.085	0.292
0.01	2.8	0.50	0.69	0.76	0.51	1.00	0.41	1.40
0.005	5.5	0.99	1.38	1.51	1.00	1.98	0.81	2.79

Although no measurements of fuel holdup appear to be available, one estimate expressed was that the holdup might be about the same as the amount of fuel injected by the accelerating pump, which is 1 to 2 cc. On this basis, it would appear that the slip factor is on the order of 0.01. Visual impressions gained from transparent-manifold movies are that a number of engine cycles occur during the interval required for a liquid-film surface wave to traverse the manifold. Since surface waves travel faster than the bulk film, visual evidence is that the slip ratio is on the order of 0.1 or less. Hence, there is reason to believe that the slip ratio is between 0.01 and 0.1.

In calculating the vaporization rate from the fuel film, use is made of Nusselt's theory of heat transfer in condensers, which predicts that the

heat transfer is controlled by conduction in the liquid film.⁽³⁰⁾ The appropriate temperature drop is then

$$\Delta T = \frac{q}{k_L A_s \delta} \quad (23)$$

The mass-transfer part of the vaporization problem can be treated by the conventional mass-transfer-heat transfer analogy⁽³¹⁾ with the result:

$$\frac{dm_L}{dt} = 5.9 \frac{h A_s}{p_a} (p_{va} - p_{vm}) \quad (24)$$

where h is the heat transfer coefficient between air and the wall.

This equation was solved for selected values of film surface temperature (which determines the fuel vapor pressure p_{va}) the resultant heat flux was calculated as $q/A_s = (\lambda dm_L/dt)/A_s$. The area A_s was taken to be the wetted surface area of the manifold passage. The heat flux was then used to calculate the temperature drop through the film, which, when added to the film-surface temperature, gave the required wall temperature. The partial pressures of the air and fuel, p_a and p_{vm} , were taken as their values when one-half of the fuel has been vaporized. The fuel was assumed to have the properties of n-heptane, and the heat-transfer coefficients and engine parameters were taken from the preceding discussion on manifold wall-to-air heat exchange. The following properties for n-heptane were used:

Latent heat of vaporization, Btu/lb	150.0
Specific gravity	0.684
Thermal conductivity, Btu/hr-ft-F	0.078
Molecular weight	100.0
Diffusivity (at 1 atm), cm ² /sec	0.008
Vapor pressure, in. Hg	
@ 40 F	0.59
60	1.19
80	1.95

Vapor pressure, in. Hg

@ 100 F	3.33
120	5.4
140	8.4
160	12.4
180	18.5
200	25.9

The results of these calculations are presented in the following tabulation:

Operating Condition		Wall Temperature Required to Vaporize an n-Heptane Film from the Manifold Walls with 0.3 of the Wall Covered, F		
Manifold Pressure, in. Hg Vacuum	Engine Fuel Flow, lbm/hr	Slip Factor		
		0.01	0.05	0.10
2	75.0	298	190	176
10	43.5	198	153	148
15	23.7	144	126	123

From these results it appears that, most of the fuel will be vaporized under typical road-load conditions, but at higher loads, increasing quantities of liquid may be present at the intake valve.

The use of a real fuel instead of heptane would result in a reduction in vaporization or an increase in the computed required wall temperature. This results from the fact that the vaporized fuel is continuously swept away from the remaining liquid by the air flow. Hence, the fuel cannot be "flashed", and the vapor pressure of the remaining liquid falls as vaporization progresses. This effect, fractionation, reduces the vaporization rate, or requires a higher temperature for an equal rate, and makes the simplified calculations optimistic. The effects of fractionation of real fuels could be incorporated into a vaporization model straightforwardly, although electronic-computer calculation would probably be required.

These results are generally in agreement with observations that the fuel is completely or nearly completely vaporized during part-load operation, and that significant portions of the fuel are unvaporized at WOT.

Liquid-Film Transport

It has been pointed out previously that the motion of the liquid film on the manifold wall is induced primarily by aerodynamic and gravitational forces. It is possible to investigate the liquid-film transport analytically by the use of the Navier-Stokes equations. Hanratty and Engen⁽³²⁾ show that, if gravitational forces are neglected and steady flow is assumed with no velocity components normal to the wall, the equations reduce to

$$\frac{\partial p}{\partial x} = \mu_L \frac{\partial^2 u_L}{\partial y^2} \quad (25)$$

With the appropriate boundary conditions,

$$u = 0 \text{ at } y = 0, \text{ and}$$

$$g_c \tau_0 = \mu_L \frac{\partial u}{\partial y} \text{ at } y = \delta$$

the following expression for the volumetric flow rate is obtained by integration:

$$Q_L = -\frac{g_c}{\mu} \frac{dp}{dx} \frac{\delta^3}{3} + \frac{g_c \tau_0}{\mu} \frac{\delta^2}{2} \quad (26)$$

This analysis is valid only for laminar flow in the liquid film with a smooth film surface. A number of experimental studies have shown that the liquid film stays smooth only with very low air velocities and that air velocities beyond some critical level induce flow instabilities in the liquid film.

Kinney, Abramson, and Sloop⁽³³⁾, investigating water-solution film cooling on the inside surface of horizontal tubes, reported observing a gradual

transition from a smooth film surface to a disturbed surface. The transition region was correlated by a single film-thickness parameter which included the wall shear stress and liquid viscosity and density.

Knuth⁽³⁴⁾, observing similar instability in the flow of water-solution films in a 3-inch plastic pipe, correlated the transition by a dimensionless film-thickness parameter and a second parameter, the ratio of the viscosity of the gas-vapor mixture to that of the liquid film.

Liquid-film thicknesses were on the order of a few thousandths of an inch in both these studies.

Hanratty and Engen⁽³²⁾ observed concurrent flow of air and water in a 1-inch by 12-inch rectangular duct with liquid-film thickness of about 0.20 inch. Several film-surface flow regimes were reported, described as

- (1) Smooth surface
- (2) Two-dimensional waves
- (3) Squalls (pebbled surface)
- (4) Roll waves (analogous to slugging)
- (5) Dispersed flow, with reentrainment of liquid droplets.

Each transition took place at increasingly higher air velocities. Flow regimes were correlated using a Reynolds number for the air and a Reynolds number for the liquid film.

Ellis and Gay⁽³⁵⁾, in studying the interfacial shearing stress between air and water at low velocities in a 1.5-inch-high duct, observed considerable mixing in the liquid film even under "laminar" flow conditions, which they presumed indicated the presence of a recirculation in the liquid film close to the wall.

Van Rossum⁽³⁶⁾ studied the concurrent flow of air and several liquids in a 6-inch by 6-inch duct and observed three flow regimes: (1) flat interface, (2) wavy interface, and (3) wavy interface with atomization from the wave tops.

Transition between the regimes was correlated in two ways: by either a Weber number or a Reynolds number, both as a function of a third parameter,

$$S = \mu_2 u_g / \sigma \quad (27)$$

Ostrach and Koestel⁽³⁷⁾ point out the lack of agreement between the various investigators on criteria for film stability and suggest that a more basic consideration of the instability mechanism is necessary. Four principal types of instability are pointed out.

These studies have produced a fairly comprehensive phenomenological understanding of the mechanisms of liquid-film flow, but they have not produced an acceptable analytical basis for predicting film velocity and thickness, even under steady-flow conditions. With the pulsating flows and fluctuating pressure gradients that occur in the intake manifold, we are even further from being able to predict liquid-film flow analytically in the induction system.

Incentives for Further Analytical Studies of Air and Fuel Transport

It has been pointed out that the physics of the air flow in the intake manifold is understood, but the computational difficulties involved in achieving a comprehensive air-flow model are great. Moreover, one of the main incentives for simulating the air flow is to provide a starting point for examining the liquid-film fuel flow. It has been pointed out that there exists no acceptable theory at present for predicting the liquid-film transport.

Further, it is known that the flow of the liquid-fuel film is inherently unsteady, which will induce time variations in air-fuel ratio, and the argument that any air-fuel stratification will induce geometric maldistribution in a branched manifold has been advanced.

It is also fairly certain that most of the fuel begins traversing the manifold in the liquid film in present systems, since the fuel droplet sizes, the air velocities, and the manifold geometry appear to be such that most of the

droplets are impacted at each bend. Consequently, the use of heated manifolds in present induction systems to vaporize a major portion of the fuel is imperative if reasonably uniform distribution is to be obtained.

Since we know that the presence of an appreciable liquid-fuel film on the manifold wall is a principal cause of time variations in mixture ratio, and since we have no practical analytical grasp of the liquid-film transport, the most promising approach to achieving improved induction systems appears to be to design so as to minimize or eliminate the liquid film. This can be accomplished by the following:

- (1) Designing the atomization device to achieve substantially smaller fuel-droplet sizes
- (2) Designing the intake manifold so that the angle through which the mixture is turned in passing from the carburetor to the intake valve is minimized
- (3) Designing the intake manifold for low air velocities
- (4) Designing the cross-sectional geometry of the manifold passages for minimum droplet impaction.

It has been shown that, even if mean droplet sizes on the order of 10 or 20 microns can be achieved, not all the fuel droplets can be expected to follow the air perfectly; therefore, it can be concluded that the manifold design for minimum droplet impaction is important.

Presuming that downdraft carburetors continue to be used, and that low hood lines continue to be an important vehicle-styling consideration, the sharp 90-degree bend at the entrance to V-8 manifolds may be unavoidable. Therefore, the use of heat below the riser should be continued so as to vaporize the unavoidable (hopefully small) amount of impaction that occurs at this point.

If this approach is taken, further studies of the effect of unsteady air flow on droplet impaction and of the effect of passage cross-sectional shape on impaction appear justifiable.

Some further work on modeling the air flow is desirable as an aid to studying the droplet transport and for characterizing the air-flow pulsations that occur through the atomization device.

MATHEMATICAL ANALYSIS OF FUEL-ATOMIZATION DEVICES

- **Basic Atomization Mechanisms**
- **Correlations for the Prediction of Droplet Size**
 - **Ultrasonic Atomization**
 - **Spinning-Disk Atomization**
 - **Two-Fluid Atomization**
 - **Pressure Atomization**
 - **Electrostatic Atomization**

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MATHEMATICAL ANALYSIS OF FUEL-ATOMIZATION DEVICES

Atomization can be generally described as the process of breaking up a bulk liquid into droplets. Theoretical analyses of liquid atomization have been quite limited. One of the earliest and one of the few theoretical analyses of droplet formation was done by Lord Rayleigh⁽³⁸⁾, who treated the case of instability in an inviscid, incompressible liquid jet. His work shows a growing wave-like disturbance causing breakup from the jet. The action of the surrounding medium and turbulent flow in the jet were neglected; however, in most cases these are the most significant actions causing atomization. The failure of fundamental analyses to treat the complex procedures involved in atomization have resulted in droplet formation being described largely by dimensional analyses and correlation with experiments.

Basic Atomization Mechanisms

The steps involved in the atomization of the liquid are quite similar regardless of the atomization technique being employed. Of primary importance, and the first step in atomization, is the formation of instabilities on the surface of the bulk liquid and the growth of these instabilities to the point where protuberances or filaments form on the liquid surface. The mechanisms involved in the formation of the unstable liquid surface depend on the atomization technique being considered and may result from the action of a gas stream on a liquid surface, turbulence with an ejected liquid jet or sheet, mechanical vibration of the liquid, or electrostatically induced forces. The second step in atomization is a breakup of the filaments or sheets into droplets because of their natural instability and interaction with the surrounding gas. The final atomization step is the further breakup of these initially formed droplets by the action of the surrounding gas. The secondary droplet breakup occurs only if the initial droplets formed are above a certain size which is dependent on physical properties and the relative velocity between the droplets and the gas.

Correlations for the Prediction of Droplet Size

As mentioned previously, the atomization process is usually treated mathematically using dimensional analysis for each specific type of atomizer. It can be generally stated that smaller drops are formed when thin sheets, small jets, or thin liquid films are produced as the initial step toward atomization. The droplet size is generally analyzed in terms of dimensionless groups which reflect the ratio of the forces causing the liquid to resist atomization, such as surface tension, and the forces which are tending to cause the atomization.

The exact form of the correlations for predicting droplet size vary widely from one type of atomizer to another. In general, it can be concluded that analyses of the atomization process and the prediction of droplet sizes are very strongly dependent upon the atomization method used and on the specific geometry of the atomizer within any classification by method or technique.

The droplet sizes formed during liquid atomization are usually expressed in terms of the Sauter mean diameter, which is defined as the size of a droplet having the same surface-to-volume ratio as the entire spray. The use of mean-size data facilitates comparisons among atomizers and provides a convenient basis for the ensuing discussions. This limitation is not meant to imply that the droplets formed are monodispersed or even all of nearly the same size. Atomizers, in general, produce a wide range of droplet sizes in the spray. Predictions for the distribution of droplet sizes are relatively uncommon; however, methods for correlating experimental distributions of droplet sizes have been studied with some success.

Perhaps the most successful technique is that developed by Mugele and Evans⁽³⁹⁾ and based on a log-normal distribution truncated at some maximum stable droplet size. The distribution is characterized by three parameters which are determined from experimental measurements: the mean size, the maximum size, and the breadth of the distribution. Although correlation with experimental measurements is possible, predictions of the size distribution are not generally possible. One exception is for vaned, cupped, and shrouded

disks whose mean drop size is predicted in a later section by Equation (29). In this case, the geometric standard deviation is reported to be approximately equal to $3.3 (D_{vm})^{0.1}$; however, this relationship has been stated as having a low level of significance and no theoretical basis.⁽⁴⁰⁾ Because the distribution of droplet sizes is known to depend on factors in addition to the mean size and is generally beyond prediction, it seems sufficient for this study to consider only mean sizes.

Ultrasonic Atomization

Sonic or ultrasonic energy can be utilized for producing liquid atomization, and a variety of ultrasonic-atomizer designs have been devised whereby liquid fuel is fed to a vibrating solid surface.^(41,42) The basic atomization mechanism involves the formation of waves or unstable protrusions on the liquid surface with the subsequent separation of portions of these protrusions. The breakup procedure occurs when the amplitude of the waves exceeds a critical value. A theoretical analysis of this process has been performed by Peskin and Raco.⁽⁴³⁾

The development of ultrasonic atomizers has been dependent on the development of successful methods for the transmission and amplification of the ultrasonic energy and its application to the liquid. Numerous atomizer designs have evolved which are generally directed toward a specific use. For example, De Vilbiss Company produces a commercially available atomizer for generating therapeutic aerosols at a very low flow rate; a miniature fuel atomizer developed for a portable (military) thermoelectric generator has been demonstrated at about 0.3 gph⁽⁴⁴⁾; the API Oil Burner Research Program supported development of an atomizer for oil burners intended to operate at from about 0.25 to 0.7 gph^(42, 45); and a high-flow-rate atomizer has been developed by Martner.⁽⁴⁶⁾

It has been found, generally, that the mean droplet size of liquid atomized ultrasonically is inversely dependent on frequency. This frequency dependence is shown in Figure 19, which is adapted from a review of ultrasonic atomization by Doyle, Perron, and Shanley⁽⁴⁷⁾, who collected data from a variety of sources.

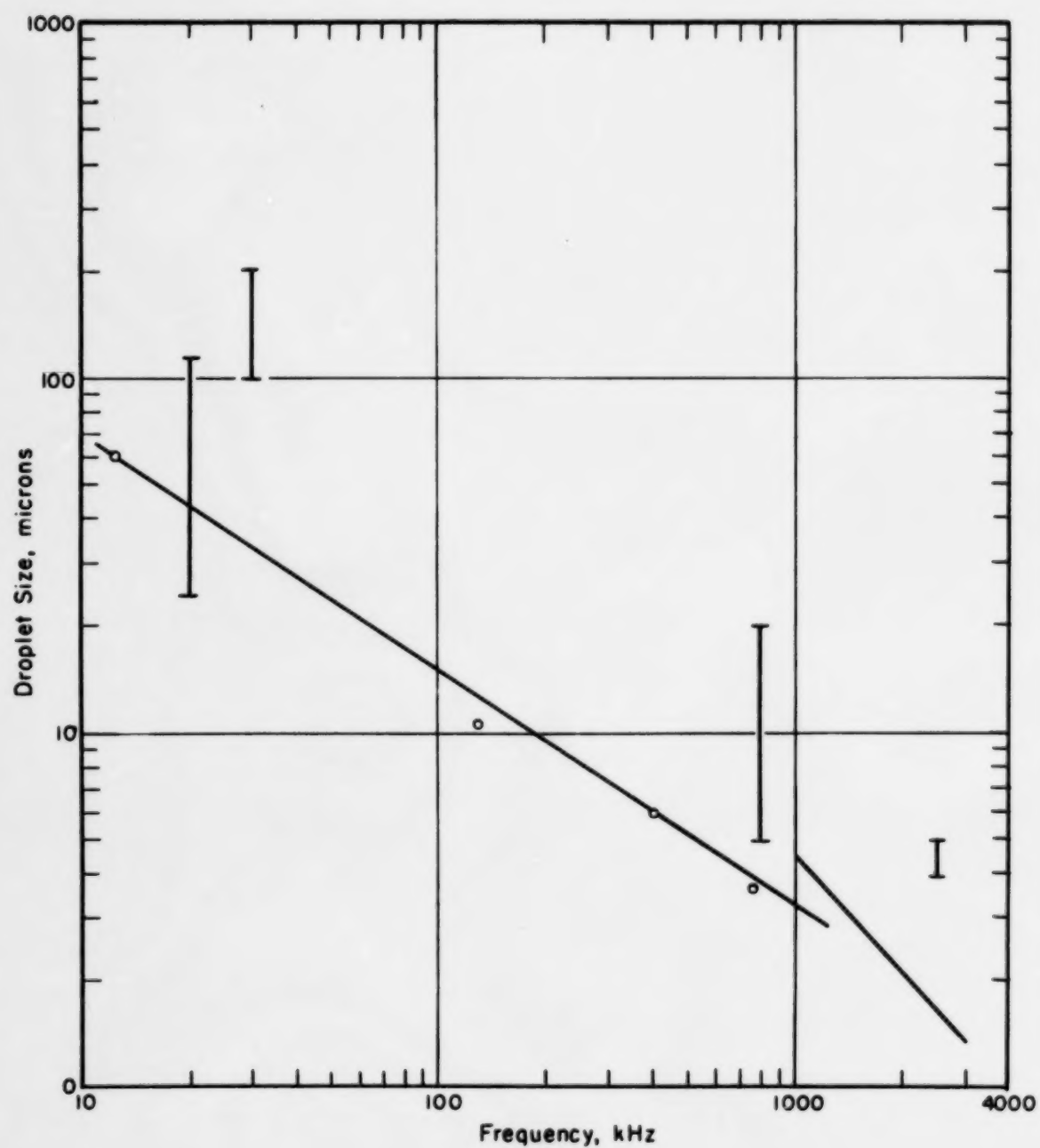


FIGURE 19. DROPLET-SIZE CHARACTERISTICS OF ULTRASONIC ATOMIZATION

It is difficult to make generalizations about ultrasonic atomization because the performance of these atomizers is quite dependent on the atomizer design and the response to varied liquid flow rate. The API atomizer referred to⁽⁴²⁾ performs quite differently depending on liquid-flow rate as a result of changes in the geometry of the liquid flow and in the excursion of the atomizing surface. Experience with ultrasonic atomizers has indicated that 25 to 100 kHz is a practical frequency range for units intended to deliver substantial flow rates, since the capacity per unit area of atomizing surface falls off as frequency is increased. According to Figure 19, an atomizer operating at 100 kHz could deliver droplet sizes in the range of 10 to 20 microns, which, according to the material on droplet transport in this report, is a potentially desirable size range. Therefore, we can conclude that the use of ultrasonic atomization in an automotive carburetor is potentially feasible, but that operation near the upper frequency limit for good flow capacity would be required.

The power required for ultrasonic atomization can be expected to vary with design, but it should be less than 10 watts for an automotive induction system. The atomizer developed by Martner⁽⁴⁶⁾ requires about 23 watts to atomize over 20 gal/hr of water.

Spinning-Disk Atomization

Spinning-disk atomization is based on the centrifugal acceleration of a liquid to a high velocity and subsequent discharge of the liquid into a gas. Although the usual terminology infers the use of a disk, there are actually several geometries which are commonly used in rotary atomizers. Perhaps the three most common types can be categorized as flat disks, cups or bowls, and vaned disks or slotted wheels. The atomization process is essentially the same for all types. Rotary atomizers operate satisfactorily under a great variety of conditions and have the advantage that the force of acceleration of the liquid can be regulated independently and, therefore, the atomization characteristics can be regulated somewhat independently from the flow rate.

Atomization from rotary atomizers requires that the liquid leaves the atomizer surface with a tangential velocity that is nearly equal to the peripheral velocity of the atomizer. For liquids having low viscosities, substantial slippage between the liquid and disk surface may occur for a flat-disk atomizer. This problem is overcome in a cup or bowl atomizer by virtue of a longer flow path which allows more time for acceleration of the liquid. The vaned-disk or slotted-wheel atomizers circumvent the slippage problem through the use of the vanes or slots.

The slippage problem for a flat-disk atomizer becomes significant when $w_L / \mu_L D_D > 4,520$ when the flow, w_L , is measured in pounds per hour, viscosity, μ_L , in centipoises, and disk diameter, D_D , in feet. For example, if a disk diameter of 4 inches is assumed and the viscosity of n-heptane is used as a reference, it can be shown that slippage does not become a problem until the flow rate is greater than 11 pounds per minute. It can be further shown that if the liquid flow rate is less than a few pounds per minute, flat-disk atomization can be effective for disk diameters of a fraction of an inch without slippage becoming significant. This would indicate that a flat-disk atomizer could operate satisfactorily for atomization of gasoline in an induction system for the liquid flow rates required.

Droplet sizes formed by rotary atomization are dependent on the specific geometry of the atomizer being used. For conditions where slippage is not a problem, the droplet sizes formed with a flat-disk atomizer are given by the following equation: ⁽⁴⁸⁾

$$D_{av} = 6.855 D_D \left(\frac{\mu_L^2}{\rho_L \sigma_d D_D} \right)^{0.041} \left(\frac{\sigma_d}{\omega^2 D_D^3 \rho_L} \right)^{0.522} \quad (28)$$

For a variety of vaned, cupped, and shrouded disks, the droplet diameter is given by:

$$D_{av} = 0.4 R_D \left(\frac{M_L}{\rho_L N R_D^2} \right)^{0.6} \left(\frac{M_L}{M_L} \right)^{0.2} \left(\frac{5.0 \rho_L L_p}{M_L^2} \right)^{0.1} \quad (29)$$

where $M_L = \omega_L / L_p$, the liquid feed rate per unit length of wetted parameter.

Figure 20 shows droplet diameters as a function of disk radius calculated for several rotation rates. These droplet sizes were calculated using the first equation above, and are for a flat-disk atomizer with n-heptane liquid. Droplet size decreases with increasing disk diameter and with increasing rotation rate. It appears that it would be difficult to utilize a disk of more than 2-in. radius in a carburetor of practical size; hence, according to Figure 20, a disk speed on the order of 30,000 rpm would be required for the generation of fuel droplets in a desired size range. Mechanically, this size-speed combination is well within the limits of conventional practice for high-speed rotary machinery. Therefore, spinning-disk atomization appears potentially usable for automotive induction systems.

Spinning-disk atomizers have the characteristic of producing nearly uniform droplet-size sprays at low liquid-flow rates; consequently, they are useful for laboratory studies requiring low droplet-size dispersion.

The energy required for atomization can be calculated as being equal to the energy to accelerate the liquid to the tangential velocity of the disk. This power for atomization is given as follows

$$E = \pi^2 R_D^2 N^2 \omega_L / g_c \quad (30)$$

It is evident that power required for atomization is proportional to fuel feed rate and the square of rotation rate and radius. Power requirements as a function of fuel feed rate are shown in Figure 21.

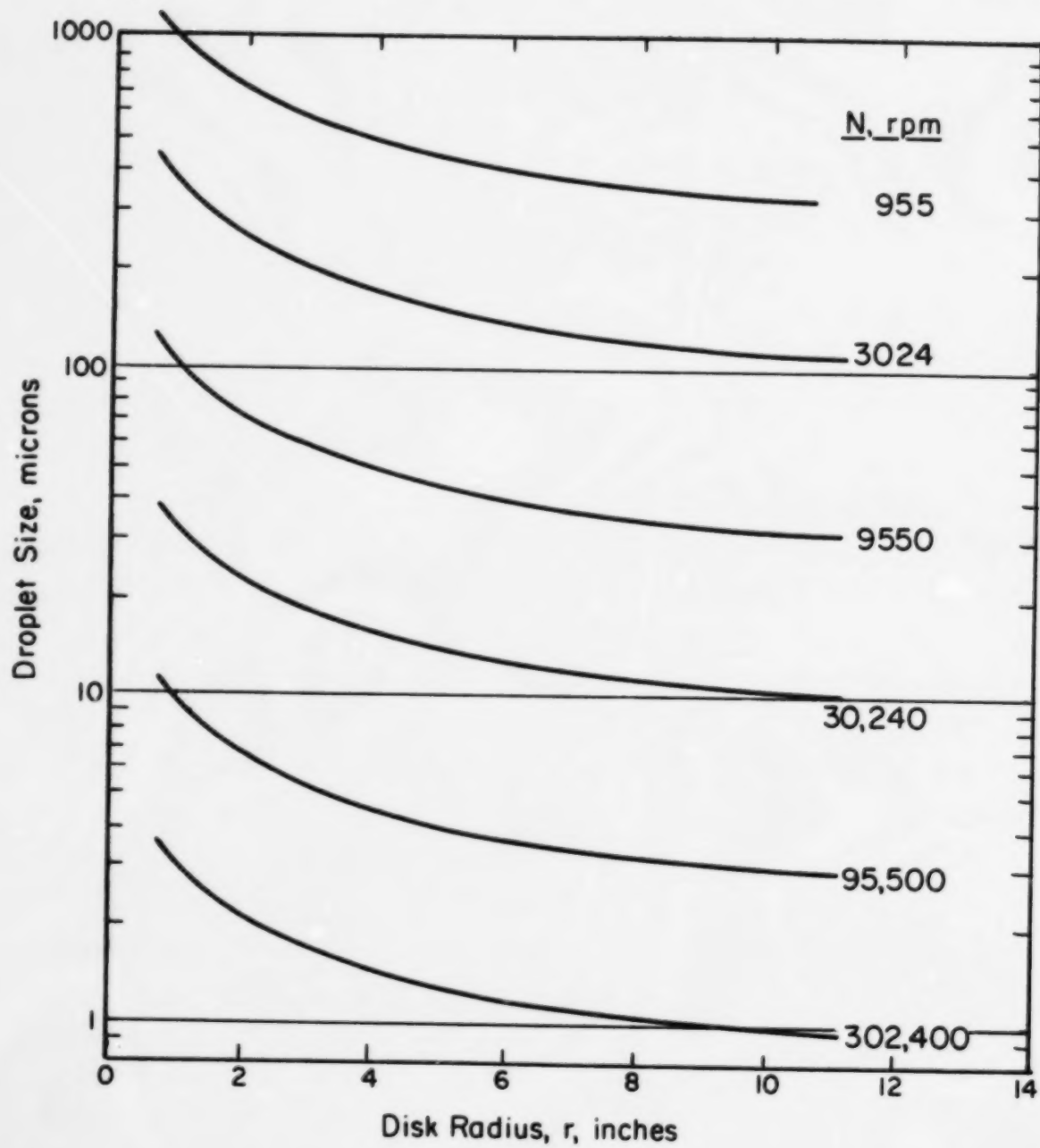


FIGURE 20. DROPLET-SIZE CHARACTERISTICS OF SPINNING-DISK ATOMIZATION WITH *n*-HEPTANE

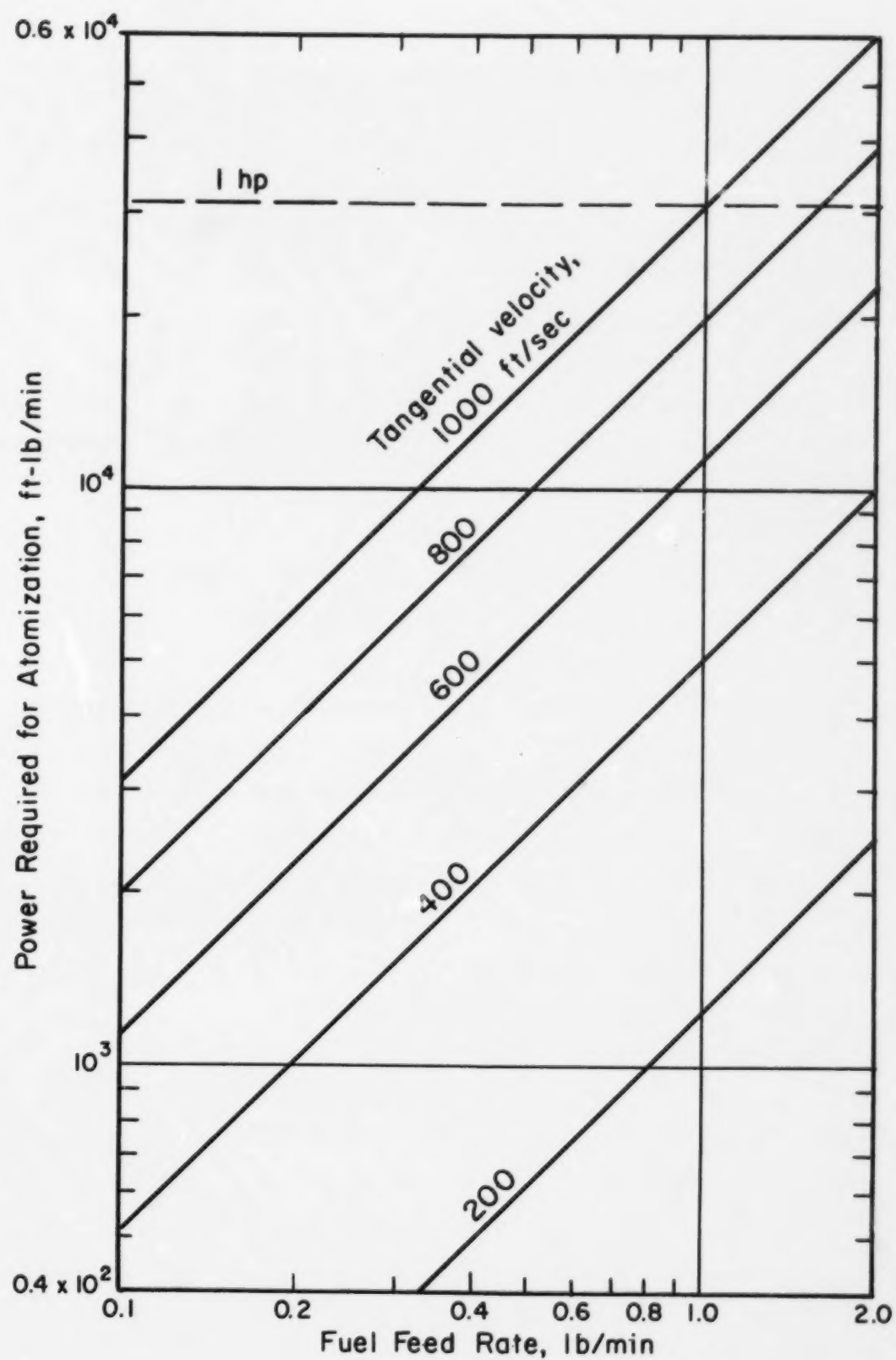


FIGURE 21. SPINNING-DISK ATOMIZATION POWER REQUIREMENT WITH n-HEPTANE

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Two-Fluid Atomization

When atomization of a liquid is brought about by the action of a gaseous medium on the liquid, the mechanism is called two-fluid atomization, or pneumatic atomization. Atomization using this technique can be accomplished in a variety of ways, but it is enhanced when the gas stream encounters an extended surface of liquid. The simplest type of two-fluid atomizer incorporates a core jet of liquid which has a high-speed gas flow directed around the outside of this liquid jet. The venturi-type atomizer used in conventional carburetors is a common type of two-fluid atomizer. A variety of air-atomization devices also can be included in the category of two-fluid atomizers.

Two-fluid atomization is generally characterized by small droplet sizes. This is because the energy contained in the gas stream can be independent of the quantity of liquid being atomized, and large amounts of energy can be supplied by the gas stream in the process of atomizing small amounts of liquid.

For liquids atomized by injection into small converging nozzles, an empirical equation predicting spray drop diameter may be written as follows:

$$D_{av} = \frac{585}{u_g} \sqrt{\frac{\sigma_L}{\rho_L}} + 597 \left(\frac{\mu_L}{\sqrt{\sigma_L \rho_L}} \right)^{0.45} \left(\frac{1000 Q_L}{Q_g} \right)^{1.5} \quad (31)$$

This relationship is the Nukiyama-Tanasawa equation⁽⁴⁹⁾; although dimensionally inconsistent, experiments have indicated that it describes adequately the droplet sizes formed with this type of atomization procedure. The equation predicts droplet size in microns and applies when the liquid density is between 0.7 and 1.2 gm/cc, surface tension between 17 and 73 dynes/cm, viscosity between 0.003 and 0.5 poise, and when the gas velocity (expressed in meters/sec) is below the acoustic level. However, experiments have indicated that the droplet-size predictions are still valid up to gas velocities of Mach 2.

The energy for atomization in a two-fluid atomizer is dependent on the design of each particular nozzle and the method of liquid injection. If a separate, high-pressure air source is used to induce atomization, the energy requirement is the equivalent of the pumping work of the air. In the case of the automotive-carburetor venturi atomizer, energy for atomization is supplied at the expense of a small loss in the total pressure of the air, which is inconsequential since no external power source is required.

Venturi-Type Atomizer. A conventional carburetor with a three-stage venturi is used in the following discussion as an example of computed performance of a two-fluid atomizer. The air velocity in the atomizing venturi was estimated using venturi-pressure curves supplied by Holley Carburetor Company. Fuel flow was computed on the assumption of a 17:1 air-fuel ratio, and the values for density and surface tension of n-heptane were used. It was found that the second term in Equation (31) was negligible for this case; thus, the droplet size is inversely proportional to the velocity in the throat of the atomizer.

Figure 22 shows that the two-fluid atomizer should be capable of producing mean droplet sizes below 20 microns with sufficiently high air velocity. By using an external atomizing air source, the droplet-size characteristics of a carburetor using two-fluid atomization could be made independent of the bulk air flow and, therefore, small droplet sizes could be obtained over the whole air-flow range of the device.

Pressure Atomization

The simplest method of atomizing liquids employs the liquid pressure as the motive force. The liquid is forced through an orifice or nozzle into the air, and instabilities in the flow of the emanating liquid cause its disintegration into a spray. Various geometries can be used for the nozzles; the simplest is merely a single jet. The shape of the jet will determine the extent of breakup of the liquid and the shape of the spray pattern. The size of the droplets formed will be dependent on the physical properties of the liquid, the diameter of the orifice, and the pressure required for the atomization. The energy required for atomization is simply the product of the pressure required for atomization and the liquid flow rate.

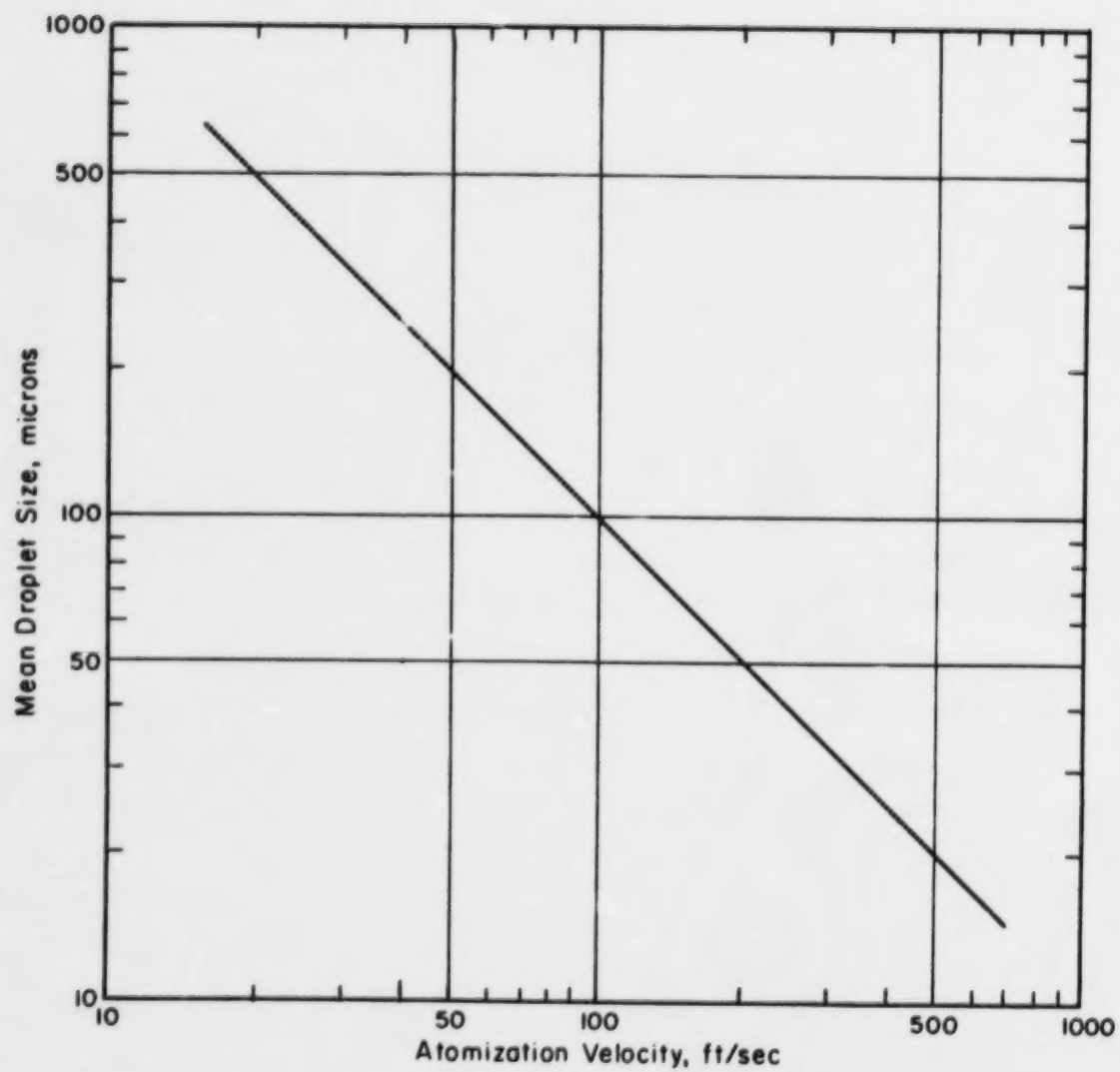


FIGURE 22. DROPLET SIZES PRODUCED BY TWO-FLUID
ATOMIZATION OF n-HEPTANE

The most widely used pressure-atomization technique employs a swirl-type nozzle. A swirl nozzle is particularly useful for low liquid flow rates and when small droplet sizes are required. In a swirl-type spray nozzle, the liquid leaves the nozzle in a sheet having the form of a hollow cone. This sheet becomes unstable as it leaves the nozzle and its breakup results in droplet formation. The conically shaped spray is achieved by the liquid swirling or rotating around the surface of a conical chamber within the nozzle. This tangential velocity plus the axial velocity results in the formation of a hollow conical sheet. The rotational velocity may be induced by the liquid entering the nozzle chamber tangentially or by the use of grooves in the flow path within the nozzle which cause a circular motion of the liquid as it enters the nozzle. The low liquid flow rates normally used with the swirl type of nozzle and the relatively small droplets formed seem to indicate that, among the various pressure nozzles, the swirl-type nozzle is best suited for application in an automotive induction system.

As in the case of pressure nozzles in general, the droplet size formed in a swirl nozzle is dependent on liquid-flow rate, orifice diameter, and physical properties of the liquid (such as kinematic viscosity and surface tension). Because of the large differences among nozzle designs and the lack of accuracy in generalized predictions of droplet size, it is desirable to limit this analysis of swirl nozzles to two types of commercially available nozzles. One type to be considered is a centrifugal swirl nozzle and the other type is a grooved swirl nozzle. These two types have been studied experimentally by Turner and Moulton.⁽⁵⁰⁾

Experimental measurements of droplet size from swirl nozzles have been reported for swirl nozzles manufactured by Spraying Systems Company.⁽⁵¹⁾ Turner and Moulton have reported that the geometric mean drop diameter on a volume basis is given as follows with the two types of nozzles: for grooved swirl nozzles (Spraying Systems Company 1/4 LN-type nozzles),

$$D_{vm} = 16.56 D_o^{1.52} \omega_L^{-0.444} \sigma_L^{0.713} \mu_L^{0.159} \quad (32)$$

and for centrifugal swirl nozzles (Spraying Systems Company 1/8 A-type nozzles),

$$D_{vm} = 41.4 D_o^{1.509} w_L^{-0.537} \sigma_L^{0.594} \mu_L^{0.22} \quad (33)$$

From the above equations, it is seen that the droplet size produced is roughly inversely proportional to the square root of the flow rate. Also, it is apparent that the nozzle orifice diameter strongly affects droplet size.

The droplet sizes given by the equation of Turner and Moulton are volumetric mean sizes. For purposes of comparison with other atomization techniques, these sizes can be converted to the Sauter mean or surface/volume mean size. It was found by Turner and Moulton that the spray drops formed were nearly log-normally distributed. Furthermore, the geometric standard deviation was approximately 1.6. If this geometric standard deviation is used and the drops are assumed to be log-normally distributed, the volumetric-mean diameter can be converted to the Sauter mean diameter in accordance with the procedures outlined by Irani and Callis.⁽⁵²⁾ The Sauter mean diameter is related to the volumetric mean diameter as follows:

$$\ln D_{Ar} = \ln D_{vm} + \ln^2 \sigma \quad (34)$$

Using a geometric standard deviation, σ , of 1.6, we find that the Sauter mean diameter should be equal to 1.25 times the volumetric mean diameter.

The mass flow rate delivered by the nozzles was obtained from the manufacturer's data for these two types of nozzles. The manufacturer's data were based on delivery rates of water at prescribed liquid pressures. These can be adjusted to be applicable for delivery of other liquids through use of the following general equation as given by Orr:⁽⁵³⁾

$$\frac{D_o^2 \Delta P \rho_L}{\mu_L^2} = 2.8 \times 10^6 \left(\frac{w_L}{D_o \mu_L} \right)^{2.4} \quad (35)$$

It is seen in the preceding equation that the mass flow rate delivered by a nozzle is determined by the product of the pressure drop and the liquid density.

The energy required for atomizing and delivering liquid at the various flow rates is simply given by the product of the flow rate and the pressure required for atomization. This relationship is shown as follows:

$$E = w_L \Delta p / \rho_L \quad (36)$$

Figure 23 shows droplet sizes and energy requirements calculated for various flow rates of n-heptane for the two types of swirl nozzles in a fuel-flow range normally required to be delivered by an induction system. In general, the smallest orifice size available for the centrifugal nozzle is somewhat larger than those available for the grooved swirl nozzles. The lower power required and larger mean droplet size for centrifugal-swirl-nozzle atomization are due primarily to the difference in orifice diameters. In comparing the two orifice diameters for the grooved-type nozzles, the effect of orifice size is seen to be considerably more significant in the power required than it is in the droplet diameter formed. One necessary precaution in application of swirl-type pressure nozzles is that, below a set liquid delivery rate (which is dependent on the nozzle design), the air core and conical pattern of the spray collapse and atomization is no longer effective. This problem, however, can be overcome through the use of duplex or double nozzles which allow a pressure actuated secondary nozzle to be brought into play to increase the liquid delivery rate. Duplex nozzles of this type are commonly used with turbine engines.

The droplet-size information presented in Figure 23 indicates that pressure-atomizing devices are generally not suitable for producing fuel spray having a Sauter mean droplet size much below 50 microns in diameter.

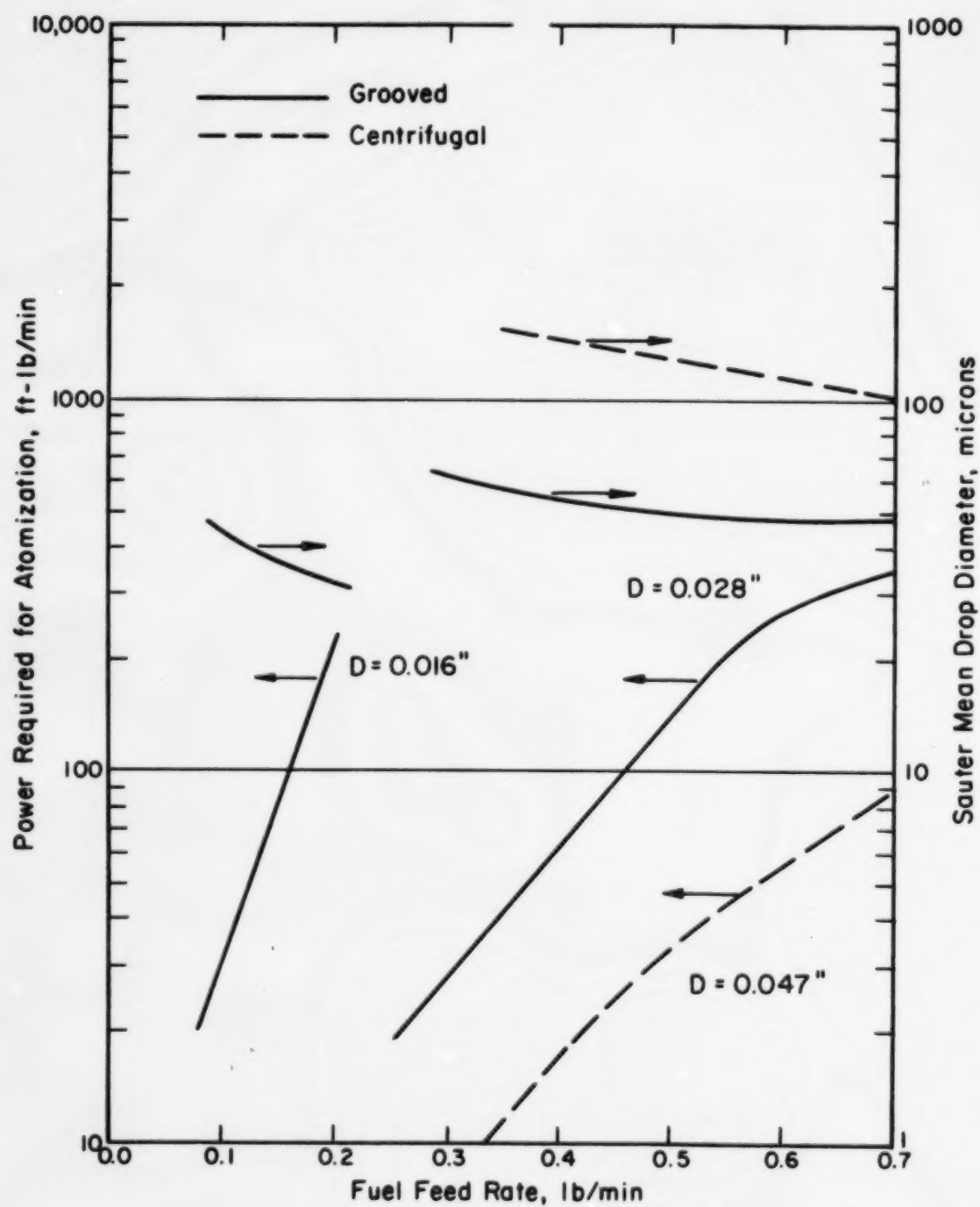


FIGURE 23. SWIRL-TYPE PRESSURE NOZZLE POWER REQUIREMENT AND DROPLET-SIZE CHARACTERISTICS WITH *n*-HEPTANE

Electrostatic Atomization

When the free surface of a liquid is subjected to intense electric fields, the liquid can spontaneously eject small droplets. The basic mechanism for producing these liquid droplets has been described by Graf⁽⁵⁴⁾, and the theory has been extended by Newgard and Noon⁽⁵⁵⁾ and by Peskin.⁽⁴³⁾

The atomization process is caused by the accumulation of a surface charge on a liquid. A portion of this surface charge (called polarization charge) is independent of the electrical properties of the liquid. Another portion of the charge (called conduction charge) is dependent on the conductivity of the liquid. The accumulation and dispersion of these charges, along with geometrical considerations such as the thickness of the liquid layer over a conductive surface, determine the characteristics of the atomization process. In general, however, the electrical charges give rise to a surface instability which is analogous to a Taylor instability. These instabilities result in the formation of liquid jets from the surface, which in turn exhibit a Rayleigh-type instability and break up into drops. The size of the droplets formed is a function of the jet radius, but, for a variety of conditions, the droplet has been found to range from 1000 microns down to submicron sizes.

Significant work on electrostatic atomization has been done under the API Oil Burner Research Program.^(43,47,55) This work has been concerned with atomization of No. 2 fuel oil for use in domestic burners. Several general conclusions have been reached which are pertinent to any application of electrostatic atomization. These include the observation that increased flow rates are always accompanied by an increase in droplet size for conditions of constant electrode voltage. In addition, it has been found that atomization-rate capability increases with increasing conductivity of the liquid, and a thin film coverage of the electrode surface enhances the atomization process. Further, the droplet size approaches a minimum and the flow rate of atomized liquid is a maximum near the voltage level that causes ionization of the air at the electrode surface.

In general, operating conditions and geometrical considerations make it difficult to achieve a flow rate much in excess of 0.05 gal/hr from a

single-ring electrode surrounding a central grounded electrode. For application to an induction system, multiple or stacked electrodes would be required to provide the necessary fuel-flow rates. In any case, the maximum flow rate achievable would be of the order of that suitable for idling or low-speed operation. For atomization of gasoline, the mass mean droplet size would probably be less than 50 microns, possibly in the range from 10 to 20 microns.

The power required for electrostatic atomization is quite small. If the energy required to form new liquid surface during atomization is taken as the reference point for determining the effectiveness of energy input into the device itself, the electrostatic process operates in the neighborhood of 30 percent efficiency as compared to 0.1 percent for a pressure-atomizing nozzle. Atomization and flow-rate capability can be improved by increasing the voltage; however, the voltage must remain below that required to cause corona.

The major areas of research necessary for application of electrostatic atomization to induction systems is the design of atomizers which provide greater capability and flexibility in flow rate and which achieve a greater degree of geometric compatibility with the system. These goals must be substantially achieved if electrostatic atomization is to become practical for use in automotive induction systems.

APPLICATION OF ALTERNATIVE ATOMIZATION METHODS FOR AUTOMOTIVE INDUCTION SYSTEMS

● General Considerations

- Fuel Metering**
- Mixture Throttling**
- Air-Fuel Mixing**

● Application of Alternative Fuel Atomizers

- The Piezoelectric Ultrasonic Atomizer**
- The Spinning-Disk Atomizer**
- The Impinging-Jet Air Atomizer**
- Swirl-Type Air-Atomizing Nozzles**
- Hartmann-Whistle Air Atomizers**

● Selection of an Optimum Atomizer

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APPLICATION OF ALTERNATIVE ATOMIZATION METHODS FOR AUTOMOTIVE INDUCTION SYSTEMS

General Considerations

It has been pointed out that conventional automotive carburetors produce sufficiently atomized fuel sprays for good droplet-transport characteristics only with the high air velocities, that occur at maximum engine power levels, and that finer atomization over the whole air-flow range can potentially be achieved by the use of one of several alternative atomization devices. Some configurations showing how several of these devices could be incorporated in an automotive induction system are presented subsequently. In addition, consideration must be given to the fact that these atomizing devices must be used in conjunction with compatible devices or systems for metering the fuel and throttling the air flow. Further, it will be important for the resulting device to achieve adequate mixing of the air and fuel.

Fuel Metering

Conventional carburetors use the venturi not only as an atomizing device but also to provide the motivating pressure for the fuel flow. The venturi acts as a fixed-area head meter, providing a pressure depression roughly proportional to the square of the volume-flow rate of gas (including air and water vapor) through the venturi. Fuel flow is metered through an orifice which passes a flow proportional to the square root of the venturi pressure drop; hence, the air venturi and fuel orifice in combination tend to deliver a constant air-fuel ratio. Compressibility effects in the air flow, temperature dependent variations in fuel density, and atmospheric variations in air density all act to cause deviations in the air-fuel metering. Temperature variations in the air are compensated for in 1969 models by the use of a thermostatically regulated air preheater using exhaust-manifold heat to maintain about 100 F inlet air temperature.

Since the venturi pressure varies as the square of the air flow, the depression at the low end of the flow range is slight, and viscous forces can cause nonideal effects on the fuel flow. In conventional carburetors, air bleed in the fuel line approaching the venturi is used to extend the practical operating range of a single venturi. However, in all conventional American automotive carburetors, a separate metering system is used for idle and low-power operation. In 4-barrel carburetors, the operating flow range of the primary venturis is further reduced by the operation of the second set of barrels at high power output.

An alternative to the fixed-area venturi that has been used in some automotive carburetion systems is based upon constant-pressure-differential, variable-flow-area operation. In such a device, the air velocity in the metering element is constant over a wide range of flow rates, and the flow area of the sensing element is varied in proportion with flow rate. This can be done by transmitting the throat pressure differential to a diaphragm control connected by a suitable linkage to an area-varying mechanism. The diaphragm adjusts the throat flow area with changes in flow rate so that the pressure drop remains constant. The position of the variable-area mechanism provides an indication of the air-flow rate which can be used, through suitable linkage with a fuel valve, to control air-fuel ratio. The air-fuel ratio can be scheduled as desired by suitable contouring of the fuel-valve characteristic relating position with fuel flow. This approach is used in the General Motors "Quadrajets" carburetor⁽⁵⁶⁾ for the high-flow section.

If a pressure-atomizing system is used, a separate fuel pump is required and the venturi pressure drop cannot be used to motivate the fuel flow; however, the venturi depression can be used as an air-flow signal in conjunction with a suitable fuel-metering device.

An alternative method of generating an engine-air-flow signal is a system which automatically computes a signal as a function of engine speed and intake-manifold air pressure and temperature. This approach is used in the Volkswagen fuel-injection system.⁽⁵⁷⁾ This requires a computer network with inputs of engine speed and of absolute air pressure and temperature in the

intake manifold. The shape of the volumetric efficiency curve and the variation of engine air flow with speed are permanently set in the computer. The computer output is a signal that is used to set the fuel rate.

To minimize the amount of development work required to obtain an operational carburetor incorporating an alternative automization device, it would be highly desirable to use an existing conventional fuel-metering system. The possibility of adapting the metering system of an existing production carburetor to the construction of an experimental induction system should therefore be considered.

Mixture Throttling

The butterfly-type throttle valve used to control the intake-manifold pressure and, hence, the engine power output performs its principal function adequately. However, it has been pointed out that the conventional butterfly valve constitutes an asymmetrical flow configuration that induces stratification of the mixture and promotes geometric maldistribution of the fuel. In addition, by virtue of its location and geometry, it provides an unnecessarily efficient target for impaction of fuel droplets. Consequently, a symmetrical, low-impaction alternative to the butterfly valve is desirable.

It has been pointed out that, in a V-8 induction system, a sharp 90-degree bend at the exit of the carburetor seems unavoidable. However, a minimum of additional flow turning at that point would be desirable for low droplet impaction. The need to provide for the vaporization of the unavoidably impacted droplets has also been discussed.

To provide both the necessary mixture throttling and flow turning, the use of a contoured plug valve appears to be a desirable alternative. The specific contour, means of actuation, and integration with the manifold inlet are design details that would need to be worked out. In the carburetor-design concepts presented subsequently, butterfly throttle plates are shown; however, it is doubtful they would be used in an induction system designed for low fuel-droplet impaction.

In systems using a variable-area air inlet and a fuel metering system based on a computed air-flow signal, it is possible to eliminate the throttle plate altogether.

Air-Fuel Mixing

The detrimental effects of stratification of the air and fuel mixture leaving the carburetor have been discussed. To promote uniform geometric distribution, a homogeneous mixture is desirable. Mixing is often accomplished in flow systems for other applications by a series of baffles. In an automotive induction system, this approach to promoting mixing is not desirable because droplet impaction would also be promoted. Accordingly, it appears that mixing can best be accomplished by the atomizing device itself through design for adequate penetration of the fuel droplets into the air stream. Concern for penetration of the fuel droplets into the air stream introduces an additional complication in the application of alternative atomization devices to carburetor design, as the geometry of the air stream and of the atomizer must be matched to obtain adequate mixing.

Application of Alternative Fuel Atomizers

In a previous section of this report on mathematical analysis of fuel-atomization devices, it has been shown that ultrasonic, spinning-disk, and two-fluid atomizers are potentially usable in automotive induction systems and that practical atomizers of the necessary flow capacity are capable of producing mean droplet sizes in the 20-micron range. High-pressure, swirl-type spray atomizers were shown to be limited to droplet sizes above about 50 microns at the moderate fuel pressure considered practical for automotive use and, therefore, less desirable for use in a low droplet-impaction system.

In the following sections, sketches are presented showing how several of the more promising atomization devices could be integrated mechanically into a carburetor--in most cases into a reasonably conventional configuration. Compatible fuel-metering systems are discussed briefly. In general, no consideration is given to details of separate idle systems or systems for maximum-power and acceleration enrichment. It is presumed that such secondary systems can be developed for each of these alternative designs using present approaches.

A piezoelectric device was selected as the most practical type of ultrasonic atomizer at present for use in an automotive carburetor, and a flat spinning disk was assumed to be the most practical rotary type of atomizer. Two types of two-fluid atomizers are treated: an impinging-jet device and a swirl-nozzle type of air atomizer. Each of these uses a substantially different induction-system configuration and fuel-metering system.

Finally, the Hartmann-whistle atomizer, which has not been discussed previously in this report, is treated briefly as an approach for obtaining exceptionally fine droplet sizes. The Hartmann-whistle can be considered a two-fluid air atomizer that utilizes ultrasonic pulsations for improved atomization at very high energy input.

In this report the performance levels and installation requirements of widely different types of fuel atomizers are illustrated. No attempt is made to identify all of the numerous atomizer designs available, nor to explore

possible combinations of principles. However, it is believed that the types discussed illustrate a range of performance levels and installation requirements appropriate to automotive application.

The Piezoelectric Ultrasonic Atomizer

In the ultrasonic atomizer, liquid fuel flows over a vibrating surface and is atomized by forces generated in the fluid by mechanical agitation.^(41,42) The fuel pressures required for atomization are negligible, so that fuel supply can be controlled as in present carburetors, using pressure drop across a venturi or other head-type metering element as the driving force. In theory, an ultrasonic atomizer could be installed within a conventional carburetor with no alteration in carburetor flow and metering characteristics. However, mechanical arrangement of the atomizer in the carburetor poses some problems.

Figure 24 shows a concept of an ultrasonic atomizer of a type that has been used for atomizing liquid fuel at rates up to about 160 lbs per hour.⁽⁴⁶⁾ An electronic power supply drives a piezoelectric disk that, in turn, causes the sharp edge of the cylindrical horn to vibrate in a radial mode. Fuel cascading down the outer surface of the horn is atomized. In this figure, fuel flows on to the cylindrical surface through a ring of small orifices placed around the atomizer.

The piezoelectric ultrasonic atomizer appears well suited to use in automotive induction systems, and it has special characteristics that must be considered in developing such an installation:

- (1) Fuel spray dispersed radially from the lower end of the cylindrical surface of the atomizer has very little initial velocity and is not projected deeply into the surrounding air stream. Consequently, dispersion throughout the air stream is dependent upon air-stream characteristics.

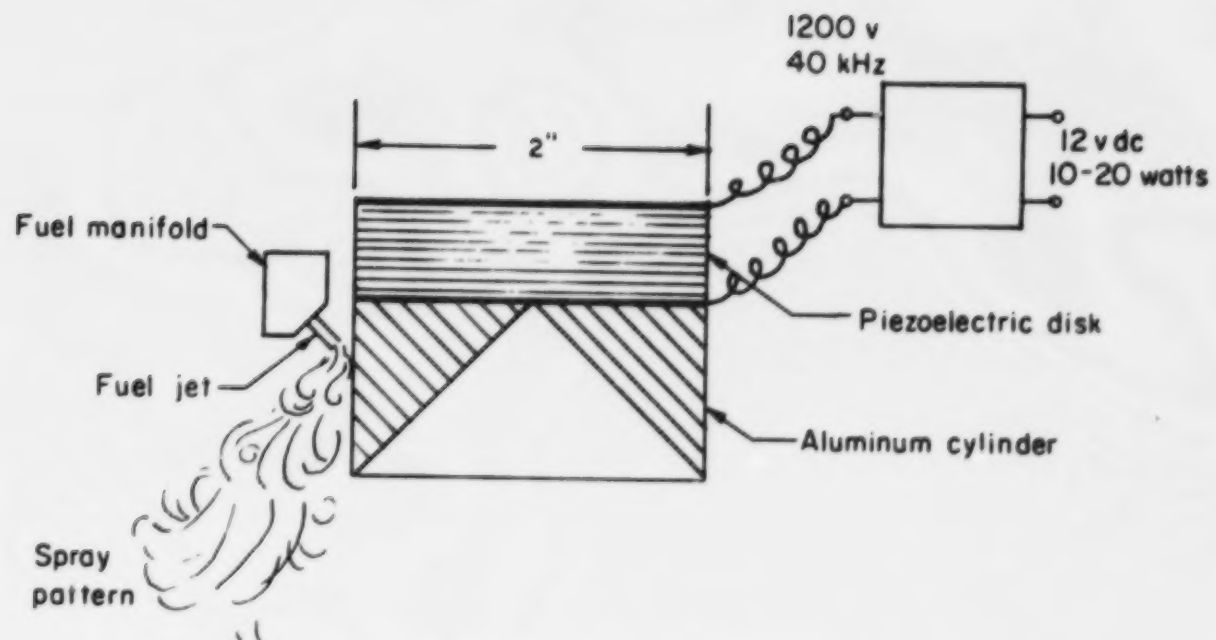


FIGURE 24. PIEZOELECTRIC ULTRASONIC ATOMIZER

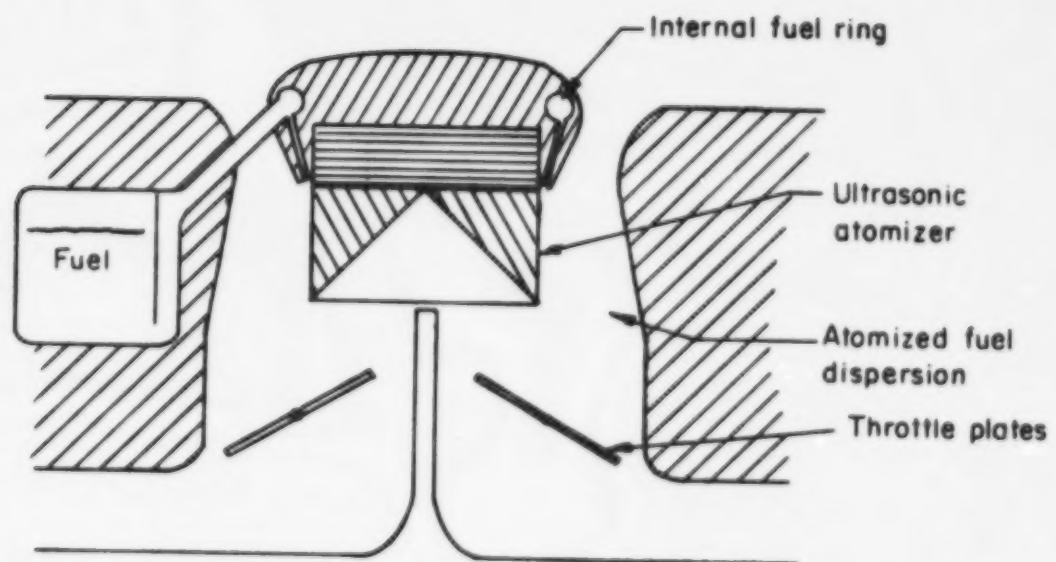


FIGURE 25. FIXED-GEOMETRY CARBURETOR WITH ULTRASONIC ATOMIZER

- (2) Small quantities of fuel jetted onto one point on the atomizer periphery will be atomized instantly at that point. Although it is usual to wet the entire surface of an ultrasonic atomizer in order to obtain maximum output, it is also possible to operate various points around the atomizer as separately controlled fuel atomizers.

Figure 25 illustrates a concept of a fixed-geometry carburetor with a piezoelectric ultrasonic atomizer at the center. In this configuration, it appears possible to carry out all carburetor functions without significant interference by the atomizer. The venturi would provide the pressure drop for fuel flow and metering as in conventional carburetors.

Figure 26 shows schematically how the ultrasonic atomizer might be used in a 4-barrel carburetor. The annular air-metering venturi is divided into two pairs of sectors, one corresponding to the low-flow venturis and one corresponding to high-flow venturis in a 4-barrel carburetor. The air passages below the atomizer merge into four throttle bores. Fuel to each atomizer sector is supplied in response to air flow through that sector, using venturi pressure drop as the driving pressure. This configuration would have the same general characteristics as a conventional 4-barrel carburetor and could incorporate the same auxiliary features to modify the air-fuel-ratio curve.

The ultrasonic atomizer also appears suitable for use with a variable-area, constant-pressure-drop air-metering system, such as that of the air-valve carburetor. This concept is attractive, as it avoids the multiplicity of passages and throttle valves needed in the 4-barrel carburetor and also provides a means of fuel metering more suitable for use with devices correcting for atmospheric density.

Figure 27 shows an arrangement of an ultrasonic atomizer in a variable-area venturi. The atomizer is part of a contoured centerbody that moves axially to adjust air-flow area so that air velocity and pressure drop are constant over the engine load range. The air-inlet passage is annular and free of flow inter-

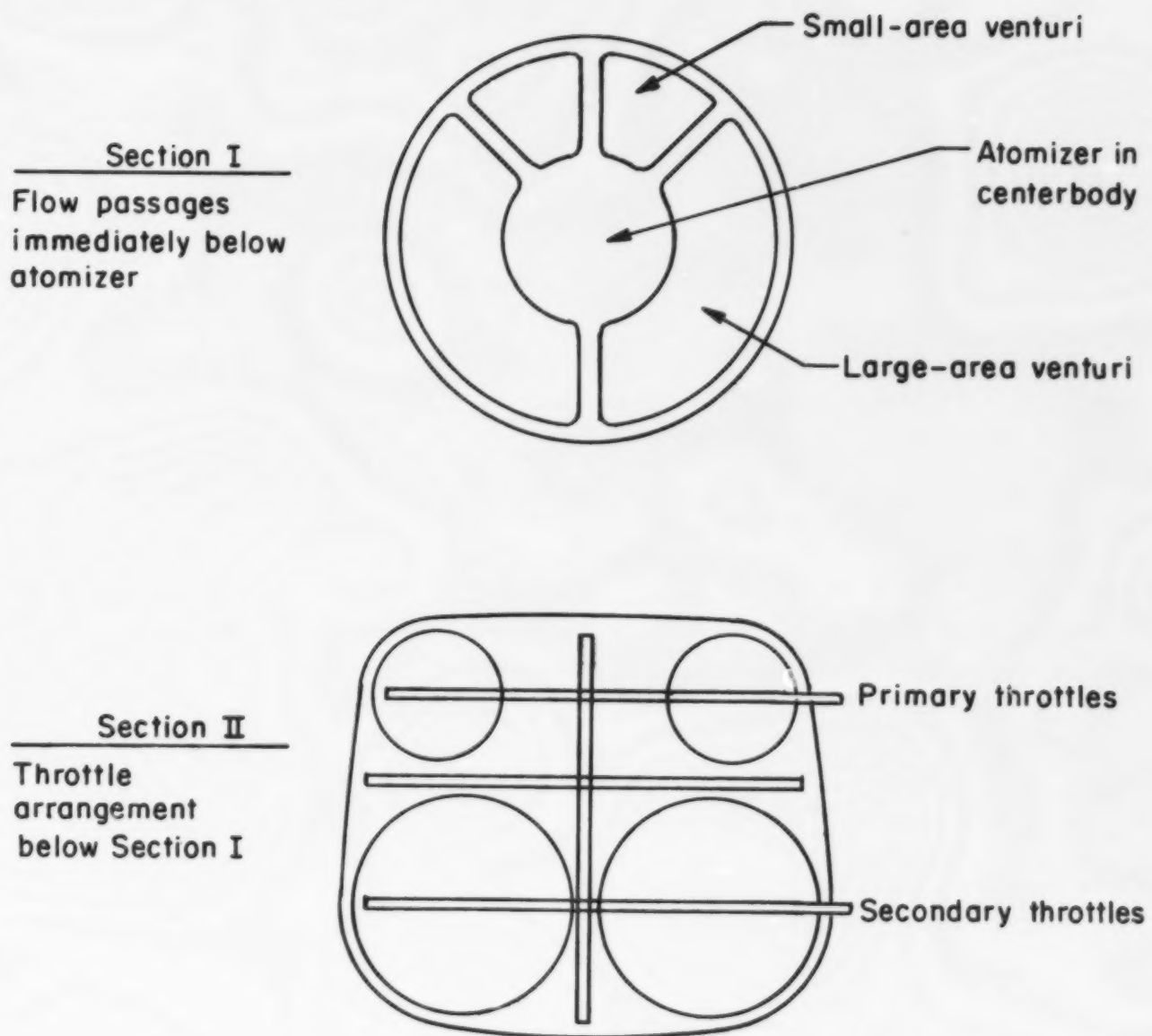


FIGURE 26. ULTRASONIC ATOMIZER IN CARBURETOR HAVING
4-BARREL FLOW CHARACTERISTICS

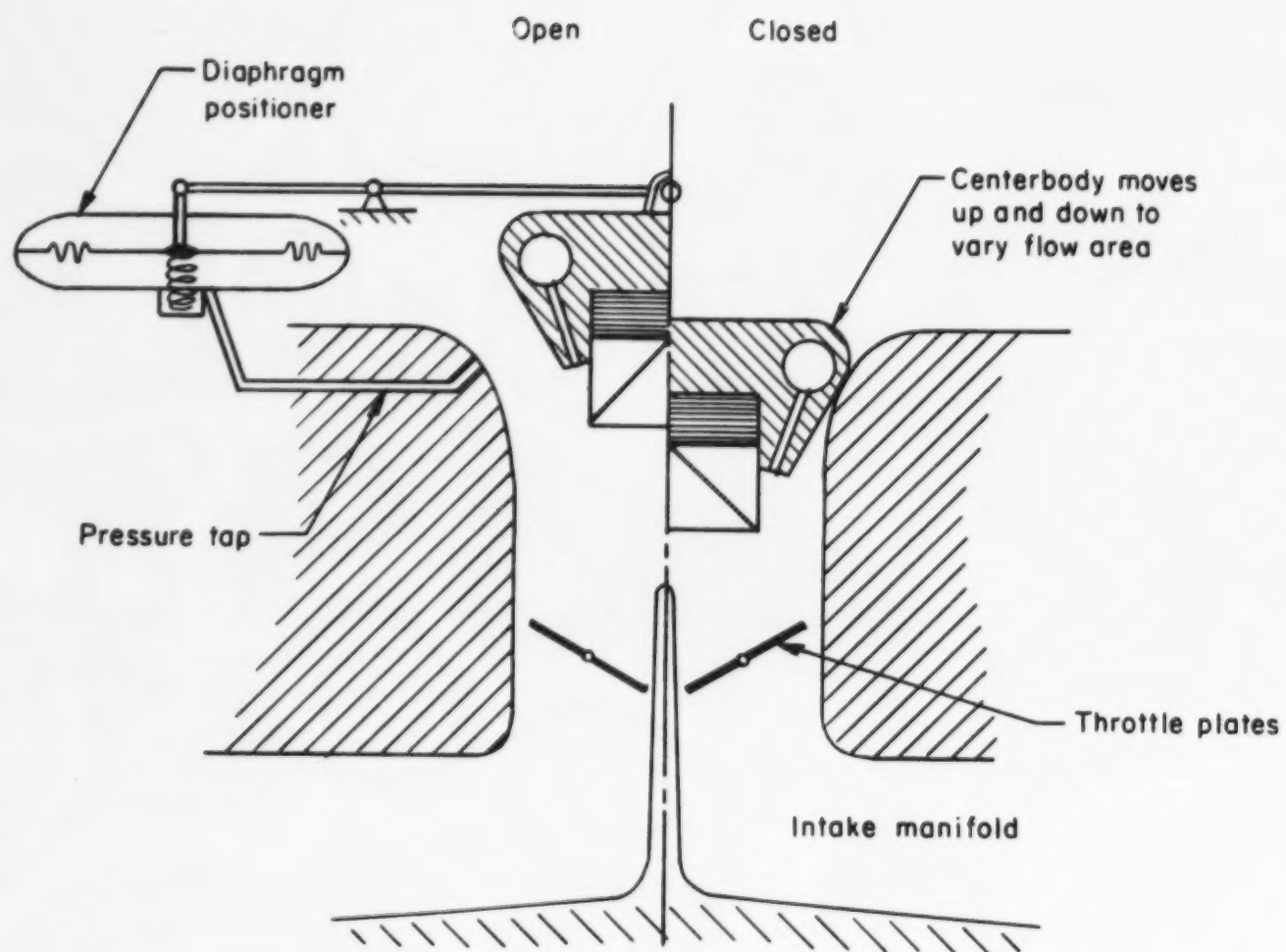


FIGURE 27. ULTRASONIC ATOMIZER IN VARIABLE-AREA, CONSTANT-DIFFERENTIAL INDUCTION SYSTEM

ference, helping to provide dispersion of fuel. The axial position of the centerbody is controlled by a diaphragm through a suitable linkage, which is illustrated schematically. This linkage would also adjust the position of a fuel-control valve to provide the desired air-fuel ratio over the load range. Fuel pressure drop is that between the venturi throat and the bowl level, as in current carburetors, and is essentially constant over the engine load range. It appears practical to incorporate additional elements in the fuel-controlled linkage to compensate for air temperature and absolute pressure, in order to provide greater precision in metering of fuel. A separate idling system consisting of separate fuel orifices and air slots or holes that remain open with the centerbody completely closed could also be incorporated for precise control of idling conditions.

It is also possible to use the ultrasonic atomizer with a fuel-metering system using a computed air-flow signal. The appearance of the atomizer and surrounding venturi would be similar to that of Figure 27, except that centerbody position could be controlled by the throttle linkage and fuel would be controlled by a computing control with inputs of manifold pressure, manifold temperature, and engine speed. For idling operation, the inlet would be closed, metal to metal, with air entering through slots or holes sized to provide the correct air flow. The throttle plates shown in Figure 27 would be eliminated.

All of the above examples appear technically feasible, and could probably be applied with sufficient development. However, it is probable that one would prove more practical than the others after initial development.

The Spinning-Disk Atomizer

The spinning-disk atomizer throws droplets of fuel from the edge of a high-speed rotating disk, with fuel supplied at the center. A speed of about 45,000 rpm is required for atomization to the level of 20 microns with a two-inch diameter disk. At some fuel rate, the disk would overload and the quality of atomization would decline, but it is feasible to stack several disks on one shaft to improve capacity, if necessary. It appears possible to provide good atomization throughout the range of fuel flow required by automotive engines.

The spinning-disk atomizer disperses spray uniformly around its periphery in a radial direction. In order to supply fuel to two venturis, it would be necessary to arrange them as sectors of an annular passage around the atomizer disk. To provide the equivalent to a 4-barrel carburetor, it would be necessary to use two disk atomizers, one for the small venturis and one for the large venturis, in order to supply each venturi with the desired fuel and air rates. Another approach would be to consolidate the flow from the two venturis on one side to a single passage in which the atomizer would be located.

Figure 28 shows a basic spinning-disk atomizer. Either an electric motor or an air turbine could be used to drive the disk.

Figure 29 illustrates a concept of a spinning-disk atomizer in a basic carburetor. The atomizer has little or no effect on air and fuel metering, so that the basic metering characteristics of the carburetor could be the same as for a standard type.

Although it appears possible to incorporate a spinning-disk atomizer in an automotive induction system, the need for a high-speed drive system complicates the design and makes it appear less promising than other types of atomizers discussed herein.

The Impinging-Jet Air Atomizer

For the purposes of this report, air atomization is discussed as a specific type of two-fluid atomizing system in which the energy of compressed air is used to atomize the fuel.

In principle, it is possible to supply sufficient energy in a small quantity of air to atomize all of the fuel needed by the engine at full load. If the atomizing air is at high pressure, near 100 psi, the quantity of air needed for fine atomization is approximately 0.2 to 1.0 lb per lb fuel, equivalent to 1.3 to 6.5 percent of stoichiometric air. Alternatively, larger quantities of air at lower pressure can be used.

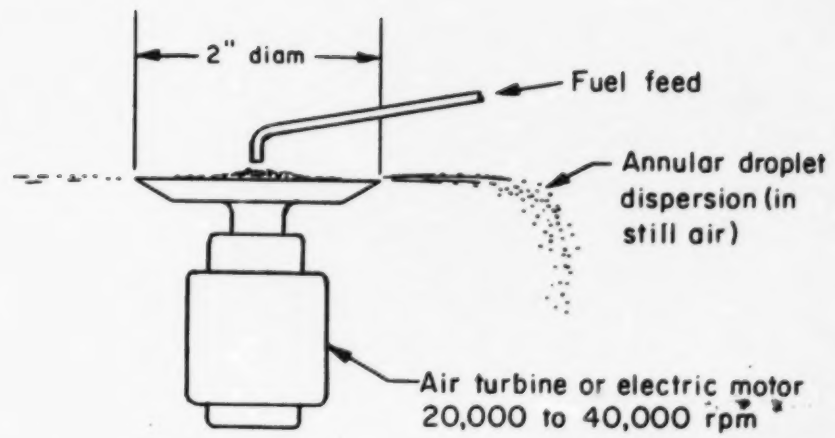


FIGURE 28. SPINNING-DISK ATOMIZER

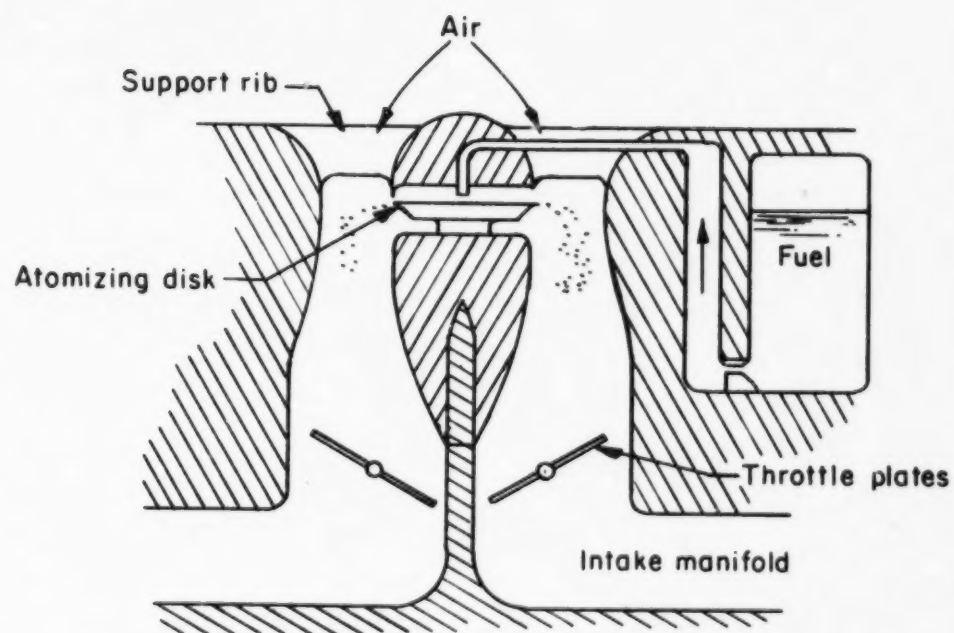


FIGURE 29. ARRANGEMENT OF SPINNING-DISK ATOMIZER IN CARBURETOR

In considering the arrangement of an air atomizer in the induction system, it is apparent that a conventional carburetor could be modified to accommodate impinging-jet air atomization with little or no change in overall arrangement or characteristics. The presence of the air jets should not influence fuel flow through the fuel orifice, as impingement takes place in the free jet away from the orifice. Since the quantity of air needed in each atomizing jet would be a small fraction of the air needed for combustion, the atomizing air would have little effect on overall air-fuel ratio, except at idle or very-low-speed operation, where a major portion of the air might be supplied through the atomizing jets.

Figure 30 shows the basic configuration of an impinging-jet atomization device. The energy in the high-velocity air is sufficient to break up the low-energy fuel jet and provide good atomization.

Figure 31 shows a possible arrangement of an impinging-jet atomizing device in a carburetor of conventional design. For each of the fuel jets in the carburetor, a high-velocity air jet is placed to impinge with it. Air could be supplied by a small compressor driven by the fan belt.

The impinging-jet air-atomization concept appears particularly attractive because it has less effect upon induction-system geometry and operation than any of the other atomizing methods that might be used. It can be used with any of the common air-metering systems applicable to induction systems.

Swirl-Type Air-Atomizing Nozzles

The swirl-type air-atomizing fuel nozzle consists of a swirl chamber in which fuel and air are mixed before entering the induction system. High-velocity air, admitted to the swirl chamber, provides the atomization energy. If nearly perfect momentum exchange between air and fuel is assumed, the atomization problem becomes one of accelerating the fuel in the swirl chamber to a velocity required for a selected level of atomization, at the expense of air momentum. It has proven satisfactory in practice to use either a small quantity

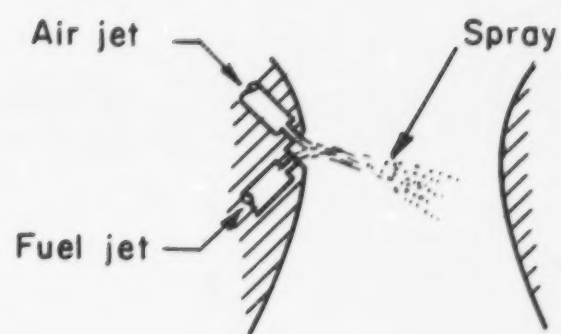


FIGURE 30. ATOMIZATION BY IMPINGEMENT OF AIR JET ON FUEL JET

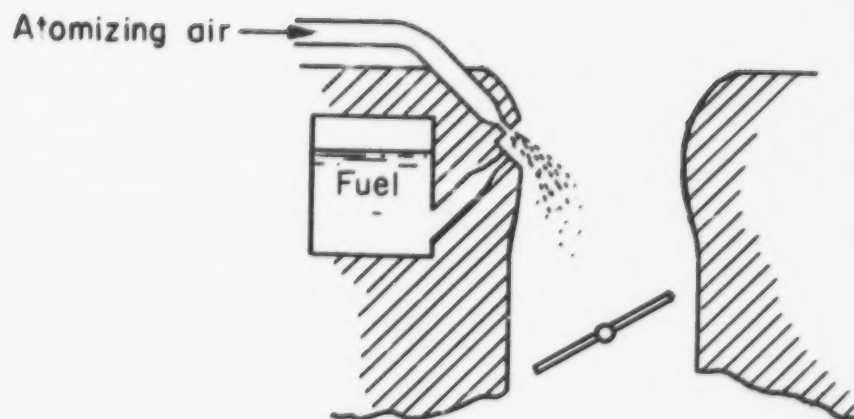


FIGURE 31. TYPICAL INSTALLATION OF ATOMIZING JETS IN CARBURETOR

of high-velocity air or a large quantity of low-velocity air as a momentum source for atomization, although the efficiency of low-velocity atomizers must necessarily be higher from fundamental momentum-energy considerations.

Figure 32 shows an induction-system design incorporating a swirl-type fuel atomizer producing a conical spray and configured expressly to minimize fuel impaction on walls of the system. In overall concept, air metering is accomplished by axial motion of a contoured centerbody via the throttle control, and the motion of the centerbody is linked to a fuel-control valve to regulate fuel-air ratio.

The outward radial momentum of the conical fuel spray can provide good dispersion of fuel in the air. The shape of the inlet bell and the centerbody would be designed to diffuse the inlet velocity and provide even distribution of the mixture flow over the cross section of the manifold to the extent possible, especially at low power levels where inlet pressure drop would be high.

The fuel atomizer would be a wide-range swirl-type air-atomizing nozzle having the desirable droplet-size distribution and spray angle. A small proportion of the total air would be admitted through an annular passage surrounding the fuel nozzle; the size of this passage would be such as to pass enough air for idling operation and atomization with the centerbody closed tightly against its seat. It is possible that the atomizing air might be sufficient for idling, in which case this bypass feature would not be necessary.

The duct immediately downstream from the inlet would be sized for moderate air velocity, and would lead directly to valve ports with the minimum possible droplet impaction on walls. The only bends in the system would be those leading directly to the intake-valve ports and, ideally, these would be contoured to provide the minimum flow-turning angle. If necessary, the manifold walls could be heated to vaporize the fuel that impacts unavoidably.

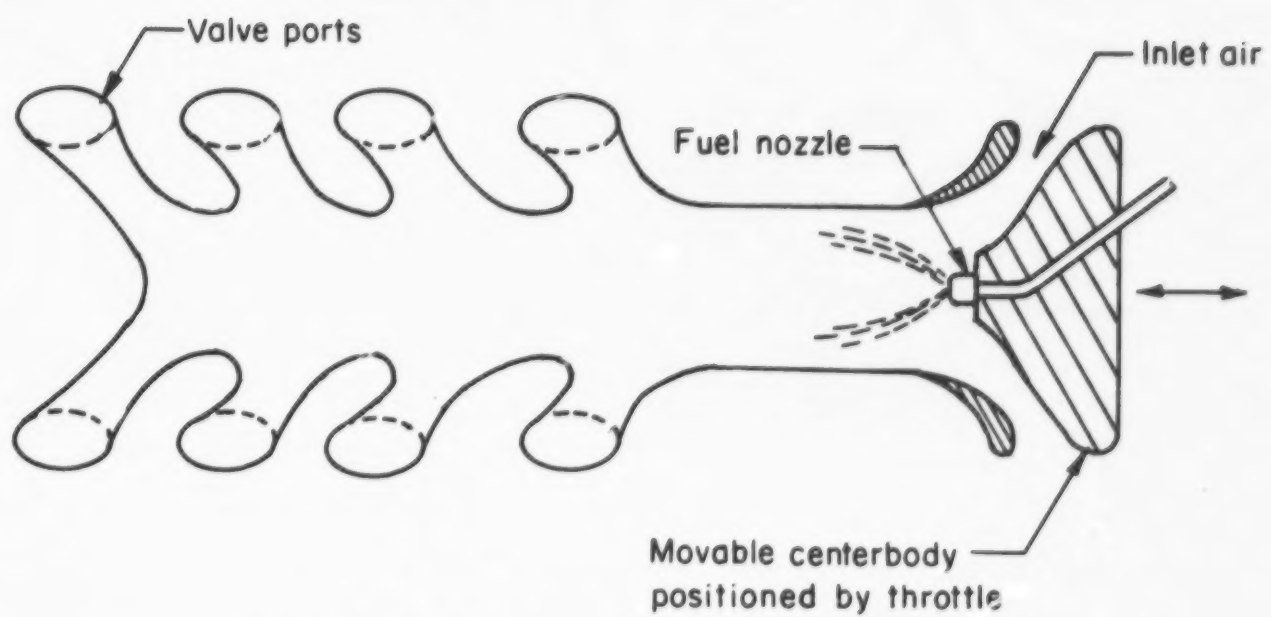


FIGURE 32. INDUCTION SYSTEM FOR SWIRL-TYPE ATOMIZERS
PRODUCING HOLLOW-CONE SPRAY

Air-atomizing nozzles have been used previously in experimental automotive induction systems. Kopa⁽⁵⁸⁾ experimented with a system in which fuel was sprayed into a heated mixing chamber. Substantial reductions in exhaust emissions with the use of this system were reported.

Although the design of Figure 32 is shown with a hollow-cone spray nozzle, it would also be suitable for any of the atomizers discussed in this report, with suitable contouring of the inlet and centerbody to accommodate the atomizer and match its spray-penetration characteristics.

A pressure-atomizing nozzle could be used should a coarser spray prove satisfactory. The details of design of the fuel pumping and control system would depend upon the type of fuel nozzle selected. A high-pressure nozzle would utilize a high-pressure fuel pump of somewhat excessive capacity at full engine rating, delivering fuel at constant high pressure by spilling excess fuel through a spring-loaded pressure-regulating valve. The high-pressure fuel would then pass through a flow-control valve consisting of a contoured needle in a nozzle driven by a linkage with the air-fuel control system. It is possible to obtain droplet sizes as small as 10 microns with fuel pressure of 1500 psi, but such pressures and the necessarily small nozzle ports do not appear practical for automotive induction systems.

Hartmann-Whistle Air Atomizers

Where extremely fine atomization is required, with mean droplet diameters below 20 microns, the Hartmann-whistle type atomizer offers promise.⁽⁵⁹⁾ In this atomizer a jet of high-velocity air is impinged upon the open end of a small cavity, which whistles with such intensity as to provide strong local shock waves in the space between the air nozzle and the cavity. Fuel introduced around the periphery of the air jet is atomized in the highly turbulent flow pattern around this cavity. The dimensions of the atomizer are usually selected such that the resonant frequency is ultrasonic, since droplet size is an inverse function of frequency and since offensive noise (possibly in the range of 100 watts of acoustic power) is minimized.

Several variations of this basic type of atomizer are available, with some using relatively low-pressure air and others using high-pressure air.⁽⁴¹⁾ The spray pattern produced is that of the turbulent jet of atomizing air heavily loaded with fuel droplets. In general, the quantity of atomizing air used would be considerably higher than for swirl-type atomizers, but the level of atomization obtained can be much finer. Of all the atomizers described in this report, the whistle-type atomizer is best suited to production of extremely fine fuel sprays at high flow rates. However, this type of atomizer is judged to be somewhat less practical than impinging-jet air atomizers for use in automotive induction systems because of its high air consumption, and would be recommended only if it proved necessary to atomize to a droplet-size range below 20 microns.

Selection of an Optimum Atomizer

Of the alternative atomization systems discussed, the piezoelectric ultrasonic atomizer appears to be the simplest. With a modified venturi section incorporating the atomizer and suitably sized and spaced fuel jets, the only additional auxiliary peripheral component is a high-frequency power supply requiring about 15 watts input at 12 volts. However, it has been pointed out that this type of device would be operating near the upper limit of its practical flow capacity if 20-micron fuel-droplet sizes were desired, so that its applicability is highly dependent upon induction-system impaction characteristics.

The necessity for a high-speed drive system is a disadvantage of the spinning-disk atomizer; in addition, it is less versatile compared with ultrasonic or impinging-jet atomizers, particularly in application to a 4-barrel unit.

The impinging-jet air atomizer requires the use of a separate air pump and modification of the fuel-metering system to account for the constant flow of atomizing air. However, the versatility, flow capacity, and atomization characteristics of this device are good. The same can be said of the Hartmann-whistle system, except that finer atomization could be obtained with greater flow of atomizing air.

The novel low-impaction induction system shown in Figure 32 with a conical-spray air-atomizing nozzle can be expected to have exceptionally low droplet-impaction characteristics and, thus, be suitable for atomizers providing 20-50 micron particle size. However, this system represents a major departure from present design and would require extensive design and development work. In addition, effects on cylinder charging would have to be evaluated.

The optimum alternative system would be, in general terms, the simplest system that can be used to achieve a major reduction in the amount of fuel flowing in the form of a liquid film on the walls of the induction system. The selection of an optimum degree of liquid-film reduction and the corresponding selection of the atomizing device can best be accomplished after further studies of fuel-droplet impaction have been completed and a firm droplet-size requirement can be established. However, at present, the jet-impinging air-atomizing system appears to be the most promising candidate.

**APPROACHES TO THE DEFINITION AND
MEASUREMENT OF MIXTURE QUALITY**

- **Significant Mixture Properties**
- **Measurement Techniques**
 - Inherent Measurement Problems
 - Fuel-Droplet Size
 - Fuel-Vapor Concentration
 - Homogeneity
 - Air-Fuel Ratio
- **Mixture-Quality Rating**
 - Relation to Mixture Properties
 - Simple Mixture Quality Tests

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APPROACHES TO THE DEFINITION AND MEASUREMENT OF MIXTURE QUALITY

In future induction-system research programs that include the development of carburetors employing alternative atomization devices and the design of low-impaction intake manifolds, it will be necessary to have available laboratory apparatus for measuring the significant properties of air-fuel mixtures. This apparatus will be useful in several ways. First, it will be necessary to evaluate the characteristics of the mixture produced by alternative atomization devices as a check on whether the desired performance is being obtained. In addition, the ability to measure certain properties of air-fuel mixtures produced in the laboratory for studies of fuel-droplet transport characteristics in various manifold geometries will be required. Also, it may be desirable in engine studies to investigate the state of the mixture as it approaches the intake valve. In these applications, the apparatus for measuring mixture properties will be used primarily as a development tool. In this report section, the properties of primary interest are identified and experimental approaches to their measurement are discussed.

In addition to investigating significant mixture properties and approaches to their measurement, the development of a mixture-quality definition and rating scheme was also the subject of one of the tasks of this program. The introduction of the concept of mixture quality is necessary if comparative evaluations of induction-system components are to be made. Approaches to the definition and measurement of mixture quality are also discussed in the following sections.

Significant Mixture Properties

In an automotive-engine induction-system study, the mixture properties of primary interest are those that affect the ability of the fuel to follow the air in the intake manifold in a uniform and steady proportion. Also important are the properties of the air-fuel mixture entering the cylinder that affect the combustion process and, hence, the exhaust emissions. However, since so little is known in regard to this second consideration, it has been considered beyond the scope of this study.

The discussion presented previously in this report in the section on mathematical approaches to intake-manifold design emphasized that the mixture properties affecting the ability of the fuel to follow the air are

- the amount of fuel in vapor form
- the amount of fuel in droplets small enough to escape impaction on the manifold wall
- the geometric homogeneity of the mixture.

It can be assumed that the remaining fuel ends up as a liquid film on the manifold wall or in larger droplets which, in turn, impact on the manifold wall. Additional mixture properties that are inherently significant are

- air-fuel ratio
- the steadiness of the mixture properties.

Steadiness in this case is taken to mean a cycle-to-cycle steadiness, or repeatability, in recognition of the fact that the mixture motion in induction systems varies cyclically.

Mixture temperatures, pressures, and velocities are secondary properties that are important only in their effect on the properties of primary importance.

Measurement Techniques

Inherent Measurement Problems

In considering techniques for measuring properties of air-fuel mixtures in an automotive-engine induction system, certain inherent measurement problems should be recognized. One of the problems is that the mixture is not in thermal equilibrium. If, for example, some of the mixture is removed from the induction system by a sampling probe, the state of the mixture at the probe exit is not necessarily the same as it was at the inlet, even if the flow of the mixture in the probe is adiabatic.

Additional problems are imposed by the unsteady flow of a multiphase mixture. One example is the difficulty introduced in isokinetic sampling of an air-droplet mixture. Isokinetic sampling is necessary if the droplet fraction is not to be biased one way or the other depending on whether the sampling velocity is greater or less than the velocity of the air-droplet mixture. In a steady-flow system, achievement of isokinetic sampling is a routine procedure, but in a pulsating flow the problem becomes very difficult. Accomplishing isokinetic sampling by matching the sampling velocity with the air velocity on a time dependent basis appears much too difficult to be practical. It seems more desirable to maintain a constant velocity in the sampling probe and correct the droplet concentration for the effects of nonisokinetic conditions. Corrections to nonisokinetic sampling have been developed⁽⁶⁰⁾ and can be applied to a problem of this type; however, prior knowledge of the droplet-size distribution is required, and the computation involved is extensive.

The cyclic variation of mixture properties and motion per se also causes certain difficulties in property measurement. Many instruments -- thermocouples, for example -- are inherently averaging devices and cannot follow the cyclic variations that occur. Any sampling probe used is likely to be a time-averaging device unless very short sampling times are used.

Consequently, only a limited amount of information on induction-system phenomena and air-fuel mixture properties can be obtained experimentally using practical instrumentation techniques and apparatus. However, it appears that this limited data will be sufficient for engineering purposes.

Fuel-Droplet Size

In the past, numerous methods have been used for measuring droplet-size distribution. A thorough review of the measurement techniques has been prepared by Norelli and Tripp.⁽⁶¹⁾ Currently, the most common methods for measuring the size and distribution of spray droplets employ photographic observation of the droplets while they are suspended in the air. Photographic techniques that have been used include not only conventional photography but holography and television scanning of the droplet distribution. The analysis of the data is done either by manual counting and sizing of the droplet images or through the use of electronic scanning equipment.

The advantages and disadvantages of the analysis techniques are fairly obvious: manual counting and sizing of the droplets is very time consuming, while electronic scanning for the same information is much more rapid but the equipment is expensive. While electronic scanning equipment is used by spray-equipment manufacturers for routine evaluation of nozzles on a high-volume basis, it is doubtful that the expense of constructing this kind of apparatus is feasible for a short-term development program. Manual counting is tedious and may require a man-day or more of effort for each test; however, it is feasible if only a few experiments are to be run.

If it is not necessary to know the droplet-size distribution, then only the fraction of droplets greater or smaller than a specified size need be measured. Measurements of this type can be made using a one-stage impactor. The general principles of impaction from curvilinear flow, as outlined in the previous section on fuel-droplet dynamics, can be employed in the design of a one-stage impactor. The impactor is typically a simple device consisting of a nozzle discharging into a cylindrical duct in which a circular target is placed.

For standard kinds of geometry, the relationship between the dimensions of the impactor, the velocity of the mixture, and the droplet-size cutoff have been investigated, and calibration of the device is not necessary. Such impactors are inexpensive and easy to fabricate.

By cascading several impactors in series, each designed to impact successively smaller droplets, a droplet-size distribution can be obtained. Construction of a cascade impactor has been discussed by Mitchell and Pilcher. (62)

The use of impaction techniques would require either that the impactor be sized to handle the entire volume of the mixture flow or that a sampling technique be used. If the sampling approach is used, the probe will have to be designed to eliminate impaction in the probe itself. This precludes the use of a curved sampling tube and would require a specially constructed duct that facilitates the use of a straight sampling probe.

Photographic techniques have the advantage of being useful for investigating time variations in droplet size. Impaction techniques, on the other hand, are essentially time averaging. Photographic techniques require both illumination and visual access; consequently, the droplet sizes produced by a carburetor, for example, could best be determined by discharging into a suitable transparent duct.

If desired, pulsating air flow could be used in the apparatus for photographic measurement. The use of pulsating flow in an impactor would make the experimental results difficult to interpret.

Fuel-Vapor Concentration

Measurement of the amount of fuel present in the mixture in vapor form can be accomplished using a sampling technique. By using a small-bore probe with a conical shield pointing upstream, the fuel droplets would be impacted by the shield. The remaining mixture would be withdrawn from a sheltered region in the wake of the shield, where no droplets of significant

size could follow the air. The air-vapor mixture could be withdrawn at any desired velocity and the fuel could be condensed in a cold trap and weighed. Alternatively, gas chromatography could be used and would have the advantage of being useful for shorter sampling periods. Pulsating flow would not be expected to cause difficulties, unless flow reversals occurred cyclically.

This technique could be used anywhere in the engine induction system where a suitable sampling port could be located. Since most vaporization would be expected to occur within the intake manifold, fuel-vapor measurements in this region would be of greatest interest.

The resulting measurements would give time-averaged data, unless techniques using extremely short sampling times could be worked out.

Homogeneity

Mixture homogeneity in a general sense, can be defined as geometric uniformity of the individual properties. In this sense, we can consider several kinds of homogeneity -- fuel-droplet, fuel-vapor, and air-fuel ratio uniformity.

The homogeneity, or stratification, of the fuel-droplet dispersion in the air stream can be determined by both photographic and impaction techniques. The photographic technique can be used to determine homogeneity by dividing the flow into arbitrary zones or streamlines and processing each zone independently. Both droplet-size dispersion and droplet population can be determined for each zone.

Traversing with a one-stage sampling impactor will yield information on the distribution of droplets smaller than a certain cutoff size.

As previously mentioned, it does not appear feasible to measure droplet-size characteristics of the mixture in the engine induction system, since special apparatus is required. However, on the engine, information on the total local flow of entrained droplets without size discrimination could be gained straightforwardly with the use of a simple sampling probe. Samples would be withdrawn with some attempt to match the sampling velocity

with the average mixture velocity; the problems involved in trying to sample isokinetically in a pulsating flow have been discussed previously. This approach would not yield quantitatively precise data on the amount of fuel in droplet form because of the nonisokinetic sampling; however, the information on the relative geometric distribution of fuel droplets should be valid.

Droplets too small to impact in the bend of the probe could be taken out by a one-stage impactor and the total liquid fuel collected could be weighed. If desired, the vapor concentration in the remaining mixture sample could also be determined by methods described previously. Thus, although it does not appear feasible to obtain complete information on the homogeneity of the mixture in an operating engine induction system, data on the relative distribution of total entrained droplets and fuel vapor can be obtained.

The cycle-to-cycle steadiness of the mixture properties has been mentioned as being important. Unfortunately, most sampling techniques do not yield this kind of information. Of particular interest would be the determination of the time variation in fuel flow and the droplet-size characteristics of the mixture produced by the carburetor under air-flow conditions that prevail during engine operation. High-speed movies are a particularly useful approach for making this determination. Light-scattering measurements also offer a high-response indication of time variations in fuel-droplet concentration.

Air-Fuel Ratio

The air-fuel ratio can be measured approximately at any point within the induction system with a sampling probe. The probe could be designed to draw quasiisokinetic samples from any point in the fuel-air mixture using methods described for the measurement of homogeneity. The major problem with this sampling technique is the uncertainty of the effects of non-isokinetic sampling on the results. Air-fuel ratio measurements obtained in this manner would give data on the airborne fuel only. The flow rate of fuel in the liquid film could be calculated as the difference between the overall fuel flow and the airborne fuel flow.

A commonly used method of obtaining the average individual-cylinder air-fuel ratio has been through the use of exhaust-gas analysis. This technique has also been used to obtain cycle-to-cycle variations in the air-fuel ratio.⁽¹⁰⁾ Exhaust gas analysis is probably the best method for obtaining the overall distribution of air-fuel ratio in a multicylinder engine manifold.

The most straightforward method for measuring the time-average air-fuel ratio at the outlet of an atomizing device (or at the inlet to a manifold) is by measuring the air and fuel entering the system. The air flow can be measured very accurately by the use of critical flow nozzles and strain-gage, diaphragm-type pressure transducers. The fuel flow can be measured by the use of orifices and a differential-pressure transducer. In the fuel-flow measurement, a series of orifices is used to cover the large fuel-flow range (up to 100:1) while maintaining a reasonably small differential-pressure range for accuracy.

Mixture-Quality Rating

For this investigation, mixture-quality rating can be defined in the broadest sense as a measure of the potential of that mixture to enable the engine to operate with minimum pollutant exhaust emissions. We know that uniform geometric distribution of air and fuel, with the absence of cycle-to-cycle variations, is a mixture-quality-dependent objective that has an important relationship to minimizing exhaust emissions. Beyond that, we know very little about the effect of the properties of the mixture entering the cylinder on exhaust emissions. Hence, at this time, we can define mixture quality generally as the potential of the mixture for promoting geometrically uniform and cyclically steady distribution of air and fuel. Consequently, the concept of mixture quality has the most meaning when applied to the mixture at the outlet of the carburetor (or the manifold inlet).

Inherent in the process of establishing a mixture-quality rating is the selection of a scale. The main requirement of the rating scheme is that a mixture that is given a rating of "high" or "good" on the scale will, in general,

promote even distribution better than a mixture rating "low" or "poor" on the scale. The scheme need not be perfect to be useful or meaningful. For example, two gasolines having the same octane-number antiknock rating will not necessarily have identical knock tendencies in a particular engine. However, it has been established that there is a strong relationship between octane number and antiknock quality; therefore, the rating scheme is useful.

There are two basic approaches that can be taken toward establishing a mixture-quality rating scheme. One is to measure the mixture properties known to be significant and relate the measurements mathematically to a rating number. The second is to devise a simple experimental procedure to measure the ability of the mixture to follow the air. Both approaches are discussed below.

Relation to Mixture Properties

The significant mixture properties have been identified, and practical approaches to their measurement are available. However, present knowledge is not sufficient to relate property values to a mixture-quality rating, although certain relationships can be hypothesized. For example, the sum of the percent of the fuel in vapor form and the percent of the fuel in droplets small enough to follow the air can be expected to be a significant parameter. A homogeneous and cyclically steady mixture having all the fuel in vapor or permanently air-borne droplets could probably be given a rating of 100 on a zero to 100 scale.

The stratification and unsteadiness of the mixture can be expressed in terms of deviation from the mean, which we will call σ_x and σ_t . A mixture-quality rating expression can then be written in the form

$$\text{Quality} = (\% \text{ vapor} + \% \text{ small droplet}) F_x(\sigma_x) F_t(\sigma_t) \quad (47)$$

where F_x and F_t are presently unknown multiplying-factor functions.

At present, knowledge of the individual mixture properties is important as a development tool, but a mixture-quality rating scheme is not. Hence, there is little incentive at present to pursue this approach to mixture-quality rating.

Simple Mixture-Quality Tests

If, in the future, it should appear desirable to have a simple, convenient method for comparing air-fuel mixtures from different atomization systems, one possibility that should be investigated is the use of a test using an impaction target in a plenum. Such a target, possibly a disk of blotter paper or filter paper, could be positioned in a plenum into which the air-fuel mixture was discharging so as to cause deposition of any fuel not in the form of vapor or sufficiently small droplets.

The measure of mixture-quality resulting from this suggested rating method would be the amount of fuel not "caught" by the target relative to the total fuel flow and the area-density pattern of the impinged fuel. The fuel "caught" could be determined by weighing the target before and after a timed test period. The area-density pattern, determined possibly by adding dye to the fuel and measuring the variation in color intensity of the wetted target surface, would be a measure of stratification.

Alternatively, the target could consist of an array of tube ends. Fuel impacted in each of the tubes would flow into a small reservoir for each tube (due to gravity, there would be no net air flow through the tubes). The collected fuel would give an indication of both the amount of fuel impacted and the asymmetry of the fuel distribution.

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REFERENCES AND BIBLIOGRAPHY

- Induction System Performance
- Mathematical Approaches to Intake Manifold Design
- Fuel Atomization
- Mixture Quality

NOMENCLATURE

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The literature reviewed during the course of this study are listed here both as direct references noted in the text of the report and as bibliography. For clarity and convenience of access, the references and other literature sources are listed in four different categories which parallel, to a certain extent, the major sections of the report. These categories are (1) induction system performance, (2) mathematical approaches to intake manifold design, (3) fuel atomization and, (4) mixture quality. In each category, the numbered references are cited in the text.

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NOMENCLATURE

A_A	Surface area
A_x	Free-flow area
c	Sonic velocity
c_p	Specific heat at constant pressure
C_D	Drag coefficient
D	Diameter
D_D	Disk diameter
D_m	Manifold-passage diameter
D_o	Orifice diameter
D_{dv}	Sauter-mean diameter (see page 80 for definition)
D_{vm}	Volumetric-mean diameter
E	Power
f	Friction factor, also frequency
F_f	Friction parameter, defined by Eq. (2d)
g_c	Conversion constant, 32.2 lbm-ft/lbf-sec ²
G	Mass velocity (mass flow rate per unit area)
h	Convective heat-transfer coefficient
k	Thermal conductivity
K	Impaction parameter, defined by Eq. (12)
L	Flow-path length
L_p	Length of wetted perimeter
m	Mass
M_L	Mass flow rate per unit length of perimeter
n	Number of revolutions

N	Rotational speed, revolutions per unit time
N_{Nu}	Nusselt number, hD/k
N_{Pr}	Prandtl number, $\mu c_p/k$
N_{Re}	Reynolds number, $\rho u D/\mu$
p	Pressure
p_{vm}	Partial pressure of fuel in the air-fuel mixture
p_{vs}	Vapor pressure of fuel at liquid-film surface temperature
q	Heat transfer rate
Q	Volumetric flow rate
λ_A	Area ratio, fraction of manifold area wetted by liquid film
λ_c	Volumetric compression ratio
λ_s	Slip factor, ratio of liquid-film velocity to gas velocity
R_b	Bend radius
R_D	Disk radius
S	Liquid-film flow parameter, defined by Eq. (27)
t	Time
T	Temperature
u	Velocity
$\bar{u}$	Average velocity
V	Volume
V_D	Cylinder displacement volume
V_h	Holdup volume
w	Mass flow rate
x	Distance along flow path
y	Distance normal to flow path
γ	Ratio of specific heats

δ	Liquid-film thickness
λ	Latent heat of vaporization
μ	Dynamic viscosity
ρ	Density
σ	Geometric standard deviation
σ_s	Surface tension
τ	Residence time
τ_o	Shear stress
φ	Velocity potential,
Φ	Flow parameter defined by Eq. (12)
ω	Angular velocity, radians per unit time.

General Subscripts

a	Air
d	Fuel droplet
g	Gas
in	In
l	Liquid fuel
m	Manifold
out	Out
v	Fuel vapor.